A SOLAR-THERMAL WATER PUMP

A thesis
submitted in fulfilment
of the requirements for the Degree
of
Masters of Engineering

in
Mechanical Engineering

in the
University of Canterbury

by
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University of Canterbury
1991

ACKNOWLEDGEMENTS

My most sincere thanks go out to Dr.J.K. Raine, Dr.A.S. Tucker and Professor A.G. Williamson all of whom in turn supervised, inspired and facilitated work on this topic.

I wish to thank M. Aldridge whose support and assistance at all times was invaluable.

Thanks are also due to all the technical staff who aided the construction and assembly of the prototype pump.

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ABSTRACT

Solar-powered water pumping is an inherently sensible proposition due to the pump being able to supply water at times when it is most in demand. In general small solar-thermal water pumps hold some promise for low technology applications in developing countries. The aim of this study was to model the characteristics of an appropriate technology water pump and build a concept-proving pump.

A detailed computer model of the pump and flat plate collector gave the ability to optimise the pump's dimensions and predict the output for any average day in any location. An exergy analysis showed that the major losses in the system were due to wasteful processes within the pump's operating cycle. The pump took the form of a 450mm diameter double diaphragm suction unit with a predicted operating efficiency of approximately 1% when operating at 68°C. The concept-proving pump was set up using compressed air and preliminary runs on pentane gave an efficiency of 0.3%. The pump takes water from 7 metres below to 1.5 metres above the pump at a rate of 3 litres every 10 seconds.

A financial analysis highlighted the fact that the solar pump has higher capital but lower running costs than some of its competitors, and as such would suit certain niche applications. Recommendations are to improve the physical lay-out of the pump and improve the design of several components to enable better performance of the pump operating on the same principle.

1 INTRODUCTION

The energy basis for society's future will rest on either solar or nuclear power. These future energy alternatives have been portrayed as the soft and hard energy options respectively. To date however, only nuclear has been used on the large scale commercially.

The current state of energy reserves and usage is summarised in this chapter and is presented in a broken-down form. This includes a summary of the myriad of forms in which solar energy manifests itself.

Solar radiation comprises direct and diffuse radiation. Combined or individually, the direct and diffuse radiation can be harnessed by a multitude of devices. Within the field of solar-thermal conversion devices there is an area dealing with small scale water pumping. Given the history of solar powered water pumps, research objectives briefly describe the initial intentions and desired outcomes for the research.

1.1 THE GLOBAL ENERGY PICTURE

Energy and its harnessing is the major distinction between our current civilisation and its predecessors. This energy distinction also holds between the developed and developing countries, to such an extent that energy consumption per capita is often used as a measure of the standard of living of a particular country.

Throughout history many cultures have been based upon different forms of energy. The Romans were dependent upon conscripted labour, who acted as the energy source of their society. In 165 AD a 15 year plague decimated the slave population and the culture plunged into a 200 year decline.

The Incas relied upon wood for fuel and in religious ceremonies. As the Inca cities started to grow, the forests receded to such an extent that the culture was decentralised, eventually falling prone to invading forces.

Our current civilisation has not yet reached the point where primary energy resources are not in plentiful supply, nor do we have to change our lifestyle or domicile to keep up with the energy flows. The current fuel reserves and rate of energy usage for the major sources of energy are depicted in Table 1.

Table 1. Energy Reserves and Consumption

Type	**Reserves** $x10^{18}$ J	**Consumed** yearly $x10^{18}$ J	**R/C** (years)	**Reference** 2
Oil	4726	127	37	3,4,5,6
Natural gas	3399	71	163	3,4,5,6
Coal	32490	198	163	4,5
Nuclear energy	3376	19	177	7
Breeder	unlimited[1]	nil		8
Hydro	48500/year[1]	22		5

1 ultimate possible
2 extreme estimates are ignored and the resulting figures are an average of relevant figures in the references quoted.
3 Tabor (1985)
4 B.P. (1990)
5 Hedley (1986)
6 LeBel (1990)
7 Goldemberg et al. (1988)
8 Cochran (1974)

The R/C ratio represents resources/consumption and is in a crude form the number of years of current usage left in the individual resource. The final ratios can be manipulated to some extent as the transformation of some reserves is possible i.e. coal to gas and gas to petroleum. The use of energy reserves is also misleading as the ultimate work that can be done by the reserve is dependent upon the conversion process. This is typified by the comparison between hydro and fossil reserves, where hydro can be converted 80% into work or 100% into heat, and where fossil fuels can only be converted typically into 35% work or 100% into heat.

Nuclear power is considered to be a non-renewable source of energy because there are no commercial breeder reactors in operation. Moreover the current trends towards inherently safe reactors is a move in the opposite direction from breeders, which have intrinsically a very high power density. There has also recently been a great decline in the manufacture of nuclear power stations, due to economic and social pressures.

It is worthy of note that as one gets towards the end of the available reserves presented in Table 1, the quality of the base product will decline, thus requiring more effort to extract useful energy and releasing more harmful pollutants into the atmosphere.

1.2 THE SOLAR ENERGY PICTURE

The sun, situated 1.495 x 10^{11} m from earth, is the original source of most of the world's current energy reserves, the major exclusions being nuclear, geothermal and to an extent tidal. The sun generates its energy by the fusion of hydrogen to form helium, giving core temperatures between 8 and 40 x 10^6 K. The surface of the sun is considerably cooler, radiating the same intensity of radiation as if it were a black body at 5762 K (Robinson, 1966).

The earth intercepts approximately 5,405,075 x 10^{16} J/year or 1,353 W/m^2 of which 32% is reflected back to space by the earth's atmosphere (albedo effect). The remaining energy is used for: generating the atmospheric pressure system 1.6%, evaporation 36%, direct sensible heating 62% and photosynthesis 0.1%.

Of the 1,353 W/m^2 of radiation at the upper atmosphere, the final maximum insolation at ground level is in the order of 1,000 W/m^2. The greatest advantages of solar radiation is that the raw material cost is nil and the conversion to heat or other forms of power has other than visually low or nil pollution potential.

The main disadvantage with solar energy is that by modern standards the 1,000 W/m^2 is a very low power density. To collect, concentrate and transport the energy to where it can be used at a much higher power density, large costs are involved. A second disadvantage of solar energy is its direct availability being limited to daylight hours and thus, for ready access, large, efficient storage systems must be used in conjunction with the collector. These storage systems need to tide the energy supply over cloudy periods and store enough energy for night time usage. These energy storage systems are currently costly and generally financially unfeasible. If such systems remain too costly, the future may see people's lives once again revolving around the energy they use, rather than the current situation where energy comes in a myriad of forms to revolve around people's lives.

The energy generated by the 2% efficient photosynthesis is roughly 10 times the amount of energy used by the human race. Human utilisation of direct and indirect renewable resources is (for other than hydro power), on a relatively small scale currently. The recent usage and availability figures are as shown in Table 2.

1.3 METHODS OF SOLAR ENERGY UTILISATION

All manner of devices have been used to convert solar radiation or its effects into useful energy. There are two main categories for the conversion devices, as they rely upon either the direct radiation for impetus or harness a by-product of solar radiation.

1.3.1 Direct Harnessing

Direct Harnessing is the more capital intensive means by which to utilise solar radiation. The expense comes from the necessity to cover a large sunlit area to capture the sparse incoming radiation which is to be converted into useful energy. Within the direct category a further subdivision into: Direct Heating, Photovoltaic conversion and Chemical conversion is possible.

Table 2. Renewable Sources of Energy[1]

Type	**Maximum producible per year**	**Annual production**	**Installed capacity**
Hydro	111600 PJ	6400 PJ	497,000 MW
Tidal	756 PJ	2.16 PJ	0.82 MW
Biomass	-	2.6 PJ	-
Wave	7.2 PJ	-	-
Wind	-	4.9 PJ	1,500 MW
Geothermal heat	-	46 PJ	12,000 MW thermal
Geothermal electric	-	54 PJ	4,719 MW electrical
Photo voltaic	-	0.017PJ	5.2 MW
Solar, Non elect. or mech (drying)	-	1.3 PJ	-
Solar-thermo electrical	-	0.018 PJ	20.8 MW
Solar - mechanical	-	0.008 PJ	-
Ocean thermal	-	nil	-

[1] All figures from World Energy Conference (1986).
$P = 10^{15}$
$M = 10^{6}$

A) Direct heating has only recently been used in passive solar building design (< 50°C). It is also possible to heat fluids via flat plate solar collectors (40 → 100°C), e.g. hot water systems. For large concentrations of radiation, temperatures of 100 to 3000°C are possible. The use of parabolic or many-directional mirrors (heliostats) achieve the energy density and temperatures to enable, on the large scale, high fluid pressures and temperatures for power generating applications, on the small scale cooking and heating food is possible.

B) Photovoltaic arrays are the most modern form of direct conversion. They convert radiation into an electric current via a silicon junction. Having been around since the 1950's (Maxwell, 1983) they are becoming ever cheaper to produce, efficiencies in the order of 8 to 15% are common, and they have a maximum theoretical possible efficiency of 23%.

C) The chemical process is most commonly used by plants in the photosynthesis process, they are generally between 0.2 and 5.6% efficient (Johansson, 1978). This process is essentially the basis of all life on earth, it is at the very base of the food chain and is also the indirect producer of the fossil fuels which are currently being consumed. This complex natural reaction produces biomass and is not used in any human made reactors.

1.3.2 Indirect Harnessing

The indirect consequences of radiation are the most commonly exploited form of renewable resources for the production of useful energy. This is due to the radiation having already been collected and converted into a form which has a greater power density than the original radiation.

The major indirect methods of conversion are: Hydro, Wind and Ocean Thermal.

A) Hydropower is the most prolific and thus historically the most economic form of indirect harnessing of solar energy. Precipitation is collected over vast areas of elevated surfaces (hills) and is then channelled through a turbine as it descends to sea level. The conversion efficiency of water potential energy to power can reach 90%. The water's potential energy is able to be harnessed on all scales from large dams to micro hydro units. Pressure heads and flow rates are accommodated by the use of appropriate wheels or turbines.

B) Wind power has reached a point where it is possible to generate electricity commercially in wind farms (e.g. in California). The wind power systems use large propellers or vertical axis blades to canvas a large area of wind and extract kinetic energy. The most common wind plants are the high speed two blade propeller types, but different conditions may suit other varieties better. In general the power extractable from the wind increases with the cube of wind velocity, so the more windy the location the more economic the wind power will be. There is a 58% maximum theoretical efficiency for the wind to power conversion.

C) Ocean thermal systems work by using the small temperature gradient in the sea (especially near the equator) to run a heat engine type cycle. The temperature difference can be in the order of 25 K and the size of the resource is very large. Unfortunately the negative points are overpowering, being the low conversion efficiency because of the small operating temperature difference and the corrosion problems due to the plant's placement in sea water.

1.4 HISTORY OF SOLAR WATER PUMPING

In 1615 Solomon de Caux described a machine for raising water by the expansion of air in a solar heater (Pytlinsky, 1978). The aims of the initial trial users of solar energy in the 1800's were to provide high pressure steam which could be used to drive all means of mechanical devices, inclusive of water pumps. Experiments with solar boilers took place in the mid to late 1800's in Austria and France. The solar boilers produced steam at medium pressures to power printing presses, pump water and distil water, they used parabolic and conical mirrors to concentrate the radiation.

In 1855 Auteuills in France used flat plate collectors filled with an ammonia solution to provide medium pressure ammonia to displace a rubber diaphragm and pump water. The ammonia was then condensed and recycled into the collectors. Developments continued with different fluids in the collectors and took advantage of new technology and materials as they became available. In the 1950's work started on very hot air engines which were positioned at the focus of concentrating collectors, utilising the heat to power a Stirling cycle engine. The 1950's saw the debut of photovoltaic systems.

The more recent set-ups have concentrated on large, high efficiency systems in order to reduce the capital expenditure per unit of water pumped. Heliostats are currently leading this field in the economic sense. The heliostats focus the radiation onto a central receiving tower where the high temperature and radiation fluxes lead to high efficiency energy conversion.

The small scale water pumps have undergone less intensive development (and still work on similar principles to Auteuills' 1885 unit) but use better technology and materials.

1.5 RESEARCH OBJECTIVES

The aim of this project and thesis was to analyse and produce a simple yet effective water pump, utilising a single working fluid with which the collection of solar energy and pumping of water could be achieved.

The pump's purpose was moulded around the possible market for low head water pumps for use in rural areas in developing countries. This imposed the major constraints on the design as is discussed later in Chapter 2. It was decided that the collector details would not be a major part of this study, because their general characteristics are well known and documented. The main focus was on the novel water pump design and its need to be simple, reliable, inexpensive and yet effective at its chosen purpose.

2 REFINEMENT OF SOLAR WATER PUMPING SYSTEM OPTIONS

After giving thought to the design criteria for the simple solar powered water pump, an investigation into the possibilities for prime mover, solar collector and water pump were undertaken. The final section in this chapter deals with the operating principles and performance of the current and proposed water pumps which are described in other literature.

The end users, their environment, financial position and disposition towards solar pumps must be taken into account when designing an appropriate technology water pump. A scenario characteristic of a needy community and the pump's design characteristics to suit the end user has been suggested by previous literature. This report uses these as a base around which the pump is moulded.

2.1 DESIGN CRITERIA

Typically in the more remote areas of developing countries there is a lack of technical know-how and equipment. It is in these remote areas that solar water pumps would be of use practically, and financially at their most feasible. The reliability of such a device is of utmost importance. If a unit breaks down due to internal damage or wear it will fail to gain the confidence of its users, who may in turn go back to their old methods of water collection.

Should the pump fail, it would generally take some time to get word to a repair person. Ideally the pump should never break down. Any periodic maintenance of the pump should be able to be carried out by local, semi-skilled technicians. The simplicity of maintenance and manufacture is one of the primary concerns for hand pumps designed to be used in remote locations in developing countries (Arlosoroff et al. 1987). If a solar water pump were to be built to replace these hand pumps it also should conform to the standards of simplicity which have been set for the hand pumps.

Some organisations have tried to set up maintenance and repair schemes for the hand pumps which they supplied. The United Nations International Children's Emergency Fund (UNICEF) set up a three tier maintenance system in India (Appleton, 1983) which consisted of the village taking care of the cleaning of the pump and its surrounds. A caretaker or block mechanic is the next tier and s/he performs some preventative maintenance. The caretaker also reinforces the message to the villagers of the importance of clean sanitary drinking water. The third tier is a mobile group with appropriate tools to carry out all operations on the hand pumps. The mobile group is informed of problems by the caretaker and they can then take the necessary action. The final tier was very expensive due to the large areas of land to be covered. This system failed (World Water, 1981) due to long delays for repairs leading to a lack of confidence in and respect for, the pumps. The block mechanics with further education and experience started their own repairs before the mobile team arrived (Gray, 1984). This lead to a single tier maintenance structure where individuals are trained and given responsibility for their own pumps. The training enabled them to carry out repairs and maintenance but such could only work successfully with pumps designed specifically for this purpose.

The need for simple hand pumps led to the Village Level Operation and Maintenance (VLOM) scheme. The VLOM concept was initiated by the United Nations Development Programme (UNDP) and the World Bank to decentralise and reduce the costs of maintenance for the purpose of providing sanitary conditions and drinking water. The period from 1981 to 1990 was declared International Drinking Water Supply and Sanitation Decade (IDWSSD), with predictions of 20,000,000 hand pumps being needed to supply the desired water (World Water, 1981). In order to test current pumps in common circumstances and provide feedback to manufacturers on design and reliability problems (Reynolds et al., 1987), the Consumers Association Testing and Research (CATR) Laboratory was commissioned to test the pumps supplied and report back with developments needed to supply a pump to meet the criteria set (CATR, 1984). This work was to be followed up in the second phase (1987-91) with the completion of research and development, promotion of pilot plants, assistance to governments and continued testing.

The VLOM guidelines suggested were:

A) Ease of maintenance - the maintenance of pumps should require only local caretakers with minimal training, few tools and no lifting gear or vehicles to carry tools or pump components.

B) Robustness - the pump must be able to withstand: abuse, vandalism, attention from animals, climatic conditions, corrosion and sand laden water.

C) Locally Manufacturable - this ensures availability of parts and also aids the developing country into manufacturing. The pumps must be easy to manufacture from common materials. Good quality control and quality assurance for the purchaser must also be ensured. The formation of joint ventures between developed and developing countries in the manufacture of the pumps would be beneficial to both countries in terms of flows of technology and experience.

D) Standardisation between pumps is to be encouraged to reduce training for the caretakers and enable standard parts with similar dimensions to give economics of scale in manufacture. Standardisation would also ensure interchangeability between pumps.

E) Cost - the pumps should have low capital and recurrent costs.

F) Discharge Rates - the discharge rate should be appropriate for the effort put in and not need excessive effort.

For solar powered pumps for developing countries, extra criteria have been suggested by Speidel (1978):

* as high an efficiency as possible,
* self priming abilities,
* freedom from maintenance,
* simple construction,
* ease of repair,
* fabricated in user's country.

Boldt (1978) suggested:

* without moving parts,
* low import share (increases local acquaintance with automatic devices).

Amor et al. (1991) suggested:

* equivalent work to hand pump which it would replace,
* appropriate technology,
* use of a non-CFC working fluid,
* use of working fluid at above atmospheric pressure.

It has been suggested by Howes (1984) that there is a need for small solar powered water pumps for sanitary village water supply purposes. The solar powered pump could replace a hand pump or be used in areas where the water table cannot supply a large volume of water in one place but would withstand several small units in a moderate proximity to one another.

The above rough design guidelines are combined with a typical trial situation to provide a basis for the solar-powered water pump to be designed in this study. A typical situation is where the unit must be able to provide a suction lift from 0 to 8 m, with the possibility of discharge at a small height above the pump unit, it must also provide a discharge rate of 20 to 30 litres per minute (as is considered appropriate for a hand pump with such a lift (Arlosoroff et al., 1987)).

2.2 PRIME MOVERS

There are many quirks of nature which can be induced by changing a material's temperature or internal energy. Many of these quirks can be utilised by choosing an appropriate geometry or process to produce some form of motion, which in turn, can be harnessed by means of a transmission to perform useful work. The following is a summary of the basis of the major prime movers' operation and their applicability to the ideal solar water pump.

2.2.1 Thermal Expansion of Solids

The expansion of solid materials takes on four distinct classes, namely: thermal expansion, phase change, thermal contracting materials and pre-stressed thermal expansion devices.

A) <u>Thermal Expansion</u>

The pre-stressed thermal expansion device was presented by Beam et al. (1973), they described tests on a prototype and theoretical analyses for the prime mover. In the most simple arrangement the prime mover consists of a length of tube supported at both ends with central stress which is imposed by one of a number of means, namely: self weight, weights external to the system, pre-loaded bearings or an otherwise applied bending moment along the length of the pipe (e.g. by spring force applied through bearings), see Fig. 1. On application of heat just before the point of maximum stress, the thermal expansion produces a turning moment within the tube and the tube rotates. There is necessity for cooling on the opposite side from that of the heat application. For solar purposes a concentrating trough collector focusing on the side of the tube with cooling water (the pumped water) running through the centre would provide a suitable heating and cooling effect.

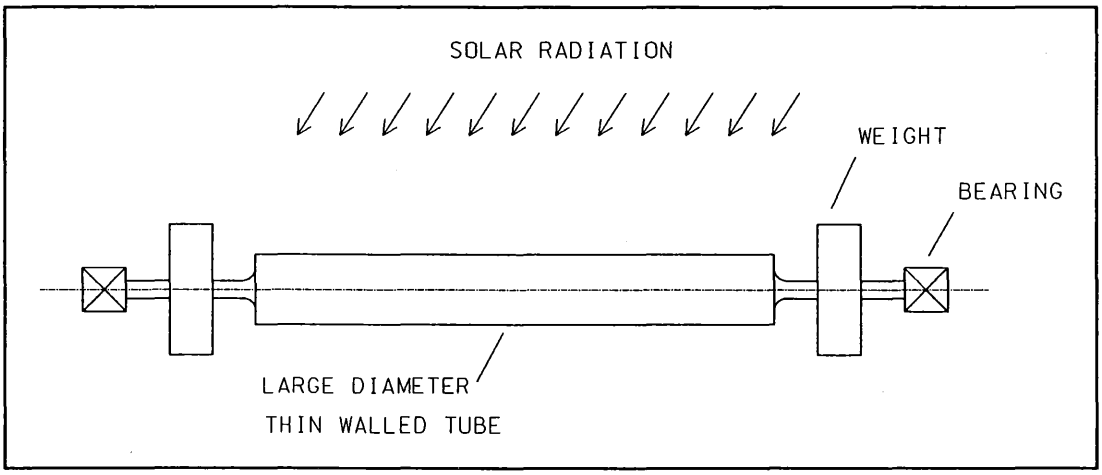

Figure 1. Pre-Stressed Tube with Weights Applying Bending Moment

The calculation by Beam et al. (1973) uses Equation 2.1 to predict the efficiency of the thermal expansion device. More complex loadings of a thermal expansion device can produce bidirectional and triaxial bidirectional stress states. The effect of these stress states is to multiply the right hand side of Equation 2.1 by 2 and 6 respectively.

$$\eta = \frac{\sigma max \ \alpha}{\rho \ Cp} \tag{2.1}$$

where:

σ max = max. stress, kPa

ρ = density, kg/m^3

C_p = specific heat, kJ/kg K

α = thermal expansion, /K

For a tube of stainless steel this equation gives 0.4, 0.8 and 2.4% for the different stress states respectively. The maximum theoretical efficiency will be substantially reduced when the maximum allowable stress is reduced to below the material's maximum stress for fatigue reasons. The rate of heat transfer and slow rotational speeds present the major problems with this prime mover.

It must be noted that Equation 2.1 is only to be taken as a rough guide to the efficiency of thermal expansion engines as Beam et al. (1973) did not analyse a full cycle for the thermal expansion engine. This oversight gives a final equation for the efficiency being independent of temperature and thus at low temperatures the thermal expansion engine's efficiency equation erroneously betters the Carnot efficiency.

B) Phase Change Materials

The discovery of the unusual properties of "NITINOL" (a Ni and Ti mixture discovered at the Naval Ordinance Laboratory in 1963) brought a rash of small engine designs. The nitinol displays a shape memory at different temperatures (Banks, 1976). The temperatures relate to a change in phase of the metal (Schetky, 1979) (Mohamed, 1979). It is possible for the metal to be trained to change shape when it moves from one phase to another. Working from theory, the maximum efficiency for two different situations have been reported. Mohamed (1979) predicts 9% efficiency when an equivalent Carnot cycle produces 20% efficiency. Tong et al. (1975) predicts 20% efficiency when the equivalent Carnot efficiency is 67%. The usefulness of such devices has been suggested by Walker (1986), to be in mainly small applications including water pumping (Banks, 1981). Many of the possible small engine configurations using shape change materials have been described by Hart (1976) and tests with real, larger scale engines have been described by Banks (1976). In general, the phase change alloys show a good fatigue life even when greatly stressed.

One of the main pitfalls of the nitinol engines is the low thermal diffusivity of the material, i.e.

Thermal diffusivity = $$\alpha = \frac{k}{\rho\, C_p} \qquad 2.2$$

where: k = thermal conductivity, kW/m K

ρ = density, kg/m^3

C_p = specific heat, kJ/kg K

For Cu, α = 1.1 cm^2/sec, Al = 0.84 cm^2/sec, Carbon steel = 0.13 cm^2/sec, Nitinol = 0.06 cm^2/sec

In order to get effective heat transfer within the nitinol, it must take the form of thin strips or wire, thus large lengths are needed to take full advantage of the phase change effects.

C) Differential Expansion Devices

Differential Expansion Devices make use of effects which can be made to happen in particular geometries where the difference in thermal expansion of two different materials produces a deflection. Such devices are normally in the form of thin sheets of two metals bonded together (bi-metallic strips) (Trihey, 1976). The most common materials are Invar and Brass. Invar is steel with 63% Nickel and has a thermal expansion of 4 x 10^{-6}/°C. It is possible to use the bi-metallic strips or sheets in place of nitinol in some of the small engines reported by Hart (1976). A bimetallic water pump has been suggested by Trihey but is a very poor and impractical design.

The efficiency of an ideal bimetallic strip has been calculated by optimising the parameters of the strip (the computer programme is included as Appendix 1). The maximum efficiency for the bi-metallic strip of length 0.13 m, with thickness of invar = 0.01 m and thickness of brass 0.005m with a temperature change of 170 K, is in the order of 0.02 % efficient.

This result is much less than the maximum efficiency which could be achieved by an individual metal expansion device, e.g. expansion of metal in general. This is because the bi-metallic strip has large internal stresses at the interface of the two metals due to their different rates of expansion, thus limiting the bending stresses and work that can be performed due to deflection of the strip.

D) <u>Thermally Contracting Materials</u>

Most solid materials when heated expand. Many metals are especially good in this respect. The main exception to this thermal expansion on heating is stretched rubber, which within a certain temperature range contracts on heating.

Rubber has a structure of multiple interwoven chains of polymer. Upon heating the stretched chains reduce in length, creating internal tension or shortening. This effect has been used in a cyclic process by Stong (1971), many other small rubber contraction engine configurations have been experimented with. The means of heating have been either hot air or direct solar radiation. Cooling of the rubber is either by natural convection or immersion in a water trough.

The maximum efficiency of these engines is greatly dependent on the rubber's properties. However in the order of 0.35% has been calculated as the highest possible efficiency (see Appendix 2).

The main limitation on the rubber engines is the size of the rubber needed to operate the engine efficiently. The thermal diffusivity of rubber is 1/25th that of nitinol (i.e. α = 0.0024 cm^2/sec); the low thermal diffusivity of nitinol was quoted as a problem, so the rubber engine is even less practical in respect to the thickness of the rubber needed for effective heat transfer. To increase the efficiency of the rubber engine it is desirable to have a large temperature difference between the hot and cold sinks and increase the strain of the rubber. Under high temperature and strain conditions the rubber is more prone to oxidation, crystallisation and creep. The rubber's perishing does not pose a large problem as rubber is cheap and easily replaced.

2.2.2 Thermal Expansion of Liquids and Gases

There are several standard cycles which rely upon expansion and contraction of fluids on either heating or vaporisation. These cycles use a working fluid which is chosen because it has the required properties for that particular cycle.

A) <u>Single Phase Systems</u>

Single phase systems rely solely on the change in volume of the fluid (gas) upon heating and include the Carnot, Stirling, Ericson, Otto, Atkinson, Brayton and Lenoir cycles. Of these, the Stirling and Brayton are in most common usage for solar applications (Capaldi, 1981). The Stirling cycle consists of four separate stages: a constant volume heat addition, a constant temperature expansion, a constant volume heat rejection and a constant temperature compression, see Fig. 2.

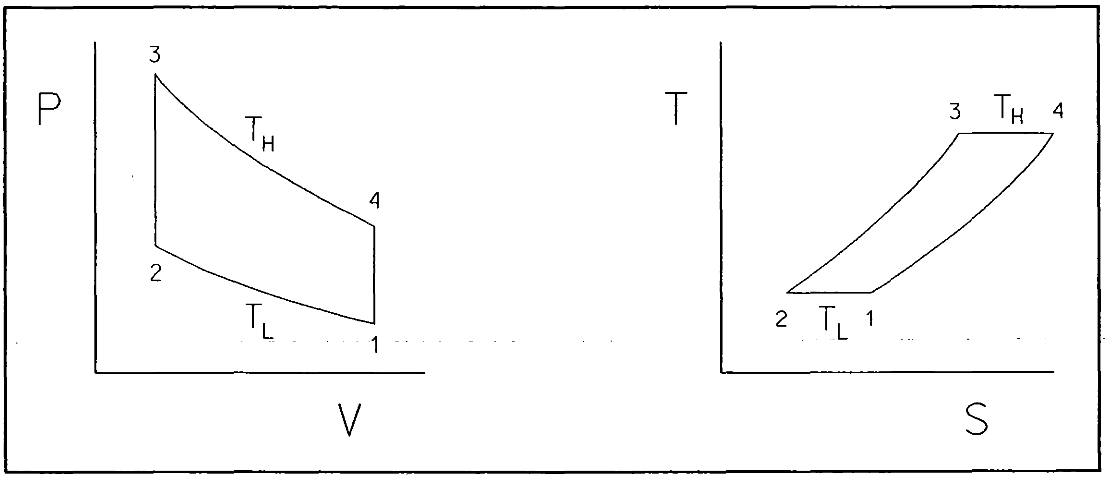

Figure 2. The Stirling Cycle

Being an externally reversible cycle the ideal efficiency will be equal to the Carnot efficiency.

The Stirling cycle has been used to power many larger scale concentrating systems and also for numerous small fluid piston type engines (Walker, 1985). The general system features a dual piston/regenerator arrangement, but can also be used in a Wankel type set-up or in an arrangement of individual pistons each 90° out of phase from the previous piston (Walker, 1978). The Stirling cycle needs a rotating output to provide the 90° out of phase needed between the piston and displacer. A degree of inertia is also needed in the shaft to ensure the whole cycle is completed.

The Brayton cycle consists of four separate processes: a constant entropy compression, a constant pressure heat addition, a constant entropy expansion and a constant pressure heat rejection, see Fig. 3.

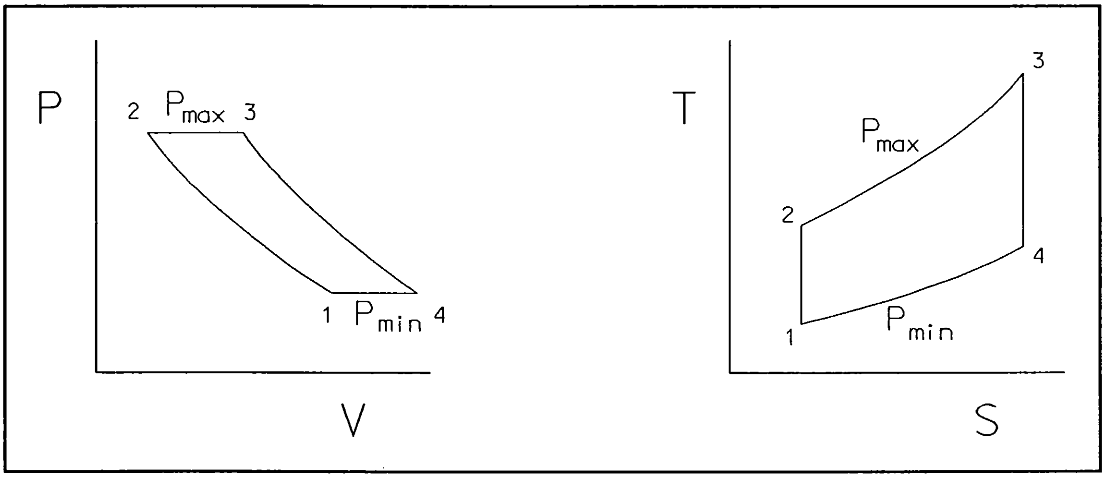

Figure 3. The Brayton Cycle

The cycle can operate as either an open or closed cycle, the former by completing the constant pressure heat rejection process external to the engine components (generally in the atmosphere). Because the heat addition occurs over a temperature range from T_2 to T_3 and the heat rejection over a range from T_4 to T_1, the Brayton cycle is inherently less efficient than a Carnot cycle operating between the same upper and lower temperature limits.

B) <u>Liquid - Vapour Systems</u>

It is theoretically possible for a unit operating wholly within the liquid - vapour region to carry out a Carnot cycle, however, for reasons of practicality the condensation process is completed, producing the Rankine cycle. The Rankine cycle and its offshoots comprise the majority of vapour cycles. The cycle consists of: a constant entropy compression, a constant pressure heat addition, a constant entropy expansion and a constant pressure heat rejection (Fig. 4). Superheating may be added to the heat addition process to enhance efficiency and reduce problems caused by excessive wet vapour in the final stages of expansion. Like the Brayton cycle, the Rankine cycle is inherently less efficient than a Carnot cycle operating between the same temperature limits. The expansion process from which the work is extracted is normally carried out in a turbine.

Of the small, less expensive engines designed to follow the ideal Rankine cycle, most fail to do so due to deviations from the ideal cycle. The deviations are generally caused by the use of ineffective components which have been designed for inexpensive manufacture (Noriaki et al., 1975).

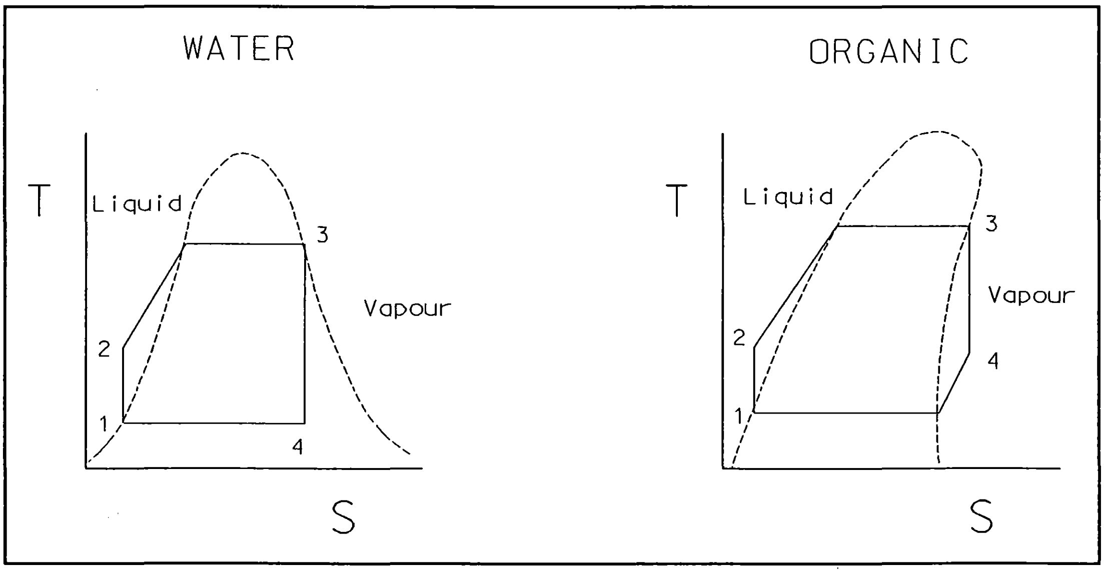

Figure 4. Rankine Cycle for Water and an Organic Liquid

2.2.3 Miscellaneous Systems

There are some thermal systems which do not fall into either of the above categories, they are powered by the difference in possible absorption by a substance of a gas or vapour at different temperatures.

A system is proposed by Schoeppel et al. (1976) where by a hydride-dehydride reactor supplies high pressure and temperature hydrogen gas to a turbine. The system relates closely to both the Brayton and Rankine cycles, with the means of compression being the main difference. The working fluid is absorbed at a low pressure and temperature, when heated it is released at a higher temperature and pressure. The absorber can be in the form of: Ti-iron alloys, Ni-Mg or Cu-Mg hydrides. The system has a flow of hydrogen from the desorption of a heated hydride bed through a turbine (extracting work) to a lower temperature bed absorbing the available hydrogen. The heat is supplied by either solar means or waste heat and is alternated with cooling to the hydride beds.

Similar absorption-desorption systems are used for refrigeration cycles (Prasad, 1986), using a weak liquor to absorb the working fluid. The stronger liquor is then heated to give off the vapour at a higher pressure and temperature. Ammonia-water or water-lithium bromide are the most common fluids.

2.2.4 Comparison of Efficiency of Prime Movers

A graph of efficiency versus temperature difference between hot and cold sides of the heat engines (Fig 5), gives a clue as to which system holds the most promise for efficient energy use. Of the cycles limited by the Carnot efficiency, the Rankine cycle has the advantage of large heat transfer being possible in the liquid to vapour stages.

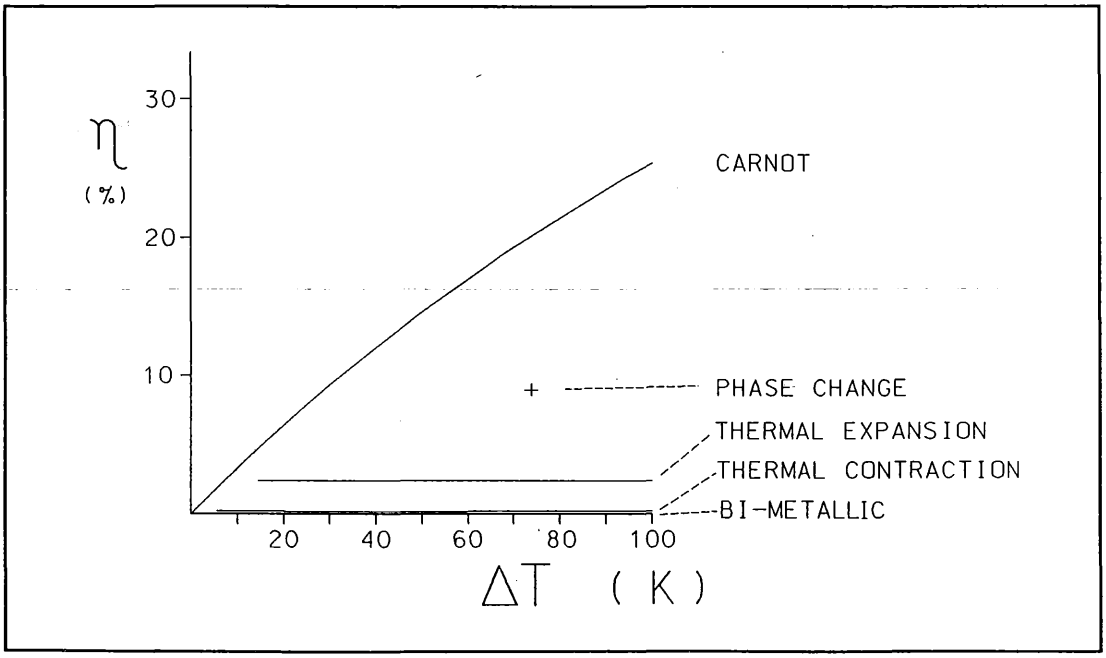

Figure 5. Prime Mover's Efficiency (T_c = 293 K)

2.3 SOLAR COLLECTORS

Solar collectors perform the function of intercepting the incoming solar radiation (photons) and degrading it into heat. There are two main categories of solar collectors. The categories are able to be separated by the degree of concentration that the radiation is subjected to before it is finally converted into heat, see Fig. 6.

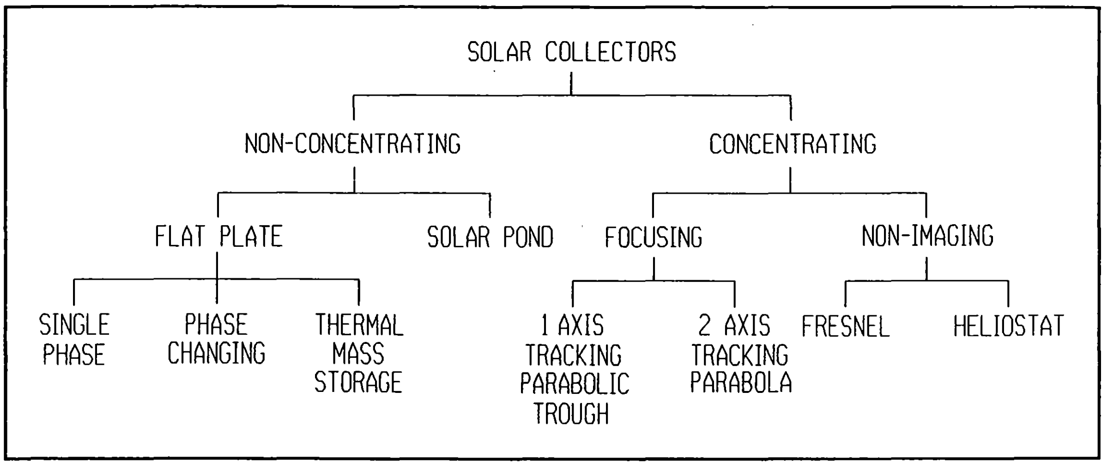

Figure 6. Solar Collector Types

2.3.1 Non-Concentrating Collectors

Non-concentrating collectors are the most common and simple group of solar collectors, and come in a myriad of forms from solar ponds and flat plate collectors through to Trombe walls. The Trombe wall and various other forms of non-concentrating collectors are designed to heat air. The air is generally moved over the darkened surface onto which the radiation is directed. The air is heated by conduction and convection and thus decreases in density. The decrease in density propels the lighter air to the top of the wall or collector, where it can be used for its desired purpose. The Trombe wall can also be used as a thermal storage medium, taking advantage of its large thermal mass.

The solar pond is a more recent advent in the solar thermal collection range and attracts attention because of its simplicity. Unlike a flat plate collector which has a panel of glazing to retard convective losses from the heated surface, the solar pond uses a large layer of water with a thermal gradient to reduce top-side heat losses. The light hitting the darkened bottom of the pond, heats the adjacent fluid. To stop this layer of heated water from rising (because of its reduced density), the pond has a salt gradient from zero at the bottom to close to saturation at the top of the pond. The salt gives a larger buoyancy gradient than that due to thermal effects and thus quashes the thermally driven convective loop. The salt gradient is maintained by the evaporation of water from the surface of the pond, increasing the salt concentration at the top of the pond. This effect ensures that the salt gradient is replenished as quickly as the general diffusion of the water can diminish it. To extract heat from a solar pond, either the layer of heated fluid at the bottom of the pond can be slowly extracted or tubes of fluid can be run through the heated layer to extract the desired energy.

The flat plate collector has been extensively studied and is a standard component for a multitude of purposes, including domestic hot water heating. The standard arrangement is to have a sheet of metal with a dark coating applied to it, to absorb the radiation. The heat is then conducted to tubes bonded to the rear of the metal sheet. These tubes contain the fluid which is to be heated. Above the darkened surface either one or two covers are placed to hinder convective heat losses to the air. The efficiency of a flat plate collector in steady state conditions is represented on a graph by the following equation (Duffie et al., 1980):

$$\eta = A - B \frac{(T_H - T_C)}{I} \qquad 2.3$$

where: A and B are constants, dependent on collector geometry and materials

T_H = Temperature of fluid leaving collector (K)

T_C = Ambient temperature (K)

I = Solar insolation (W/m^2)

graphically it takes the form of Fig. 7.

2.3.2 Concentrating Collectors

Concentrating systems generally consist of a protected reflective surface, supports, tracking systems and an absorber. The main benefits of the concentrating collectors are, firstly the temperatures achievable are much higher, leading to a greater efficiency when being used to power a Carnot type cycle. The second large advantage is that the area of heated surface is less than that over which the radiation was collected. Thus the collector has a smaller surface over which the parasitic heat loss to the atmosphere can take place.

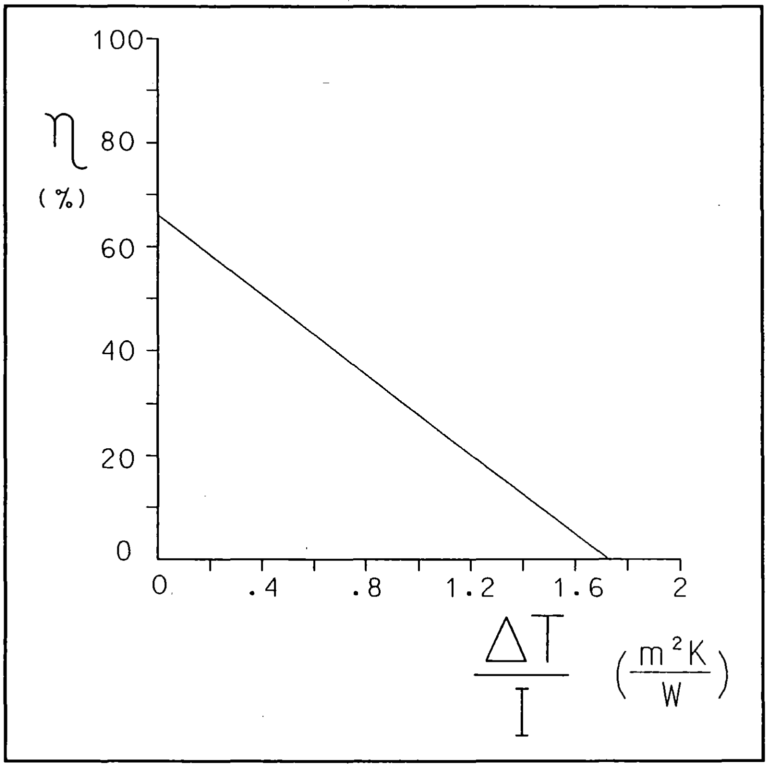

Figure 7. Collector Efficiency

For good reception of radiation over a whole day the concentrating collectors generally track the sun. For concentration factors less than 2, solar tracking is not required. Two to ten times concentration warrants one dimensional tracking with seasonal adjustments, and greater than 10 times concentration justifies full two dimensional tracking of the sun. The focusing type concentrators can be either two dimensional (line focus) or three dimensional (point focus) with concentration ratios of up to 200 and 40,000 times respectively.

No single collector unit will suit all applications so one must be chosen which fulfils the most requirements of the system, taking into account the purpose and duty of the overall unit. The characteristics of concentrating and non-concentrating collectors are compared in Table 3.

Table 3. Collector Comparison

Flat plate collectors	Focusing collectors
simple and low precision	precision built and sophisticated
low incoming energy density	medium to high incoming energy density
collects direct and diffuse radiation	collects only direct radiation
generally non tracking	tracking essential
large output variation over a day	smaller daily variation, clouds affect performance more
large thermal mass	low thermal mass
high heat losses	low heat losses
semi durable	semi durable
initial costs moderate	initial costs moderate to high
freezing can be a problem at night	much reduced susceptibility to freezing

2.4 WATER PUMPING DEVICES

Pumping devices can be divided into two main categories, either suction devices or pressure devices. The suction devices generally have all of their moving parts at the top of the well but are limited to sucking the water up to 9.5 m maximum before the pressure drop causes the water to boil (vaporise) at the low pressures. The pressure devices are typified by having most of the driving machinery at the top of the well, with either a piston, chamber or occasionally an electric motor below the water line. The below ground pump requires a large well in which to fit the below-ground components.

Within each of the above categories the pumps can be divided into broad groups depending on construction, being: centrifugal, Archimedes, chain pump, rotor/stator, diaphragm, inertia, plunger and tube diaphragm. The most common pumps in use in developing countries are currently Persian wheel for large scale pumping (500 W animal driven) or plunger type hand pumps (20 W) for the lower scale pumping.

<u>Rotating Units</u>: The most common pump for attachment to electric motors and medium speed rotating shafts is the centrifugal pump, but it is also the most unsuitable for developing countries as it must be below the water line or be primed daily before operation. It is possible to place the rotor/stator down the well but it would need a long supported rotating shaft or a waterproofed electric motor and wet bearings. Some of the latest units now have the centrifugal pump at ground level using a venturi device below the water level which incites the water to flow up the well, such systems need less bleeding. Axial flow pumps can be substituted for the centrifugal pump depending on application.

The Archimedes screw operates on a moderate tilt angle and therefore is of little use for well applications but better for open reservoir use.

The chain pump is a simple device which needs a slow rotational drive with moderate torque. The chain pump comprises numerous circular plates with sealing lips on their edges. The plates are attached at equal intervals to the chain which passes through the plate's centre. The plates are pulled through a pipe which extends from below water level to the head of the well. Some water is trapped on top of each plate. The plate empties its load at the top of the well and is recirculated to the bottom.

<u>Reciprocating Units</u>: The inertia pump operates by accelerating a mass of water upwards. While it still has upward momentum the housing surrounding the water is lowered, admitting more water into the bottom via a non return valve. When the water has lost its momentum or the housing reaches the bottom of its travel, the valve flaps shut. This gives a trapped volume of water which is once again accelerated upwards to give it upward momentum. This method is rather jerky in action with high speed and power needed for only one part of the cycle (Crouch, 1983).

A tube diaphragm pump uses above ground components to extend a tubular rubber diaphragm, which is situated at the bottom of a well and secured there by its lower end. When tension is put on the tube it elongates but contracts diametrically, decreasing its internal volume. The water held in the diaphragm is pushed through a non-return valve. On release of tension in the tube some water is sucked back into the bottom of the tube via a second non-return valve. The hysteresis in the rubber diaphragm leads to a moderately inefficient pump.

The flat diaphragm pump undergoes the same cycle as the tube diaphragm in so far as pushing the water through two non-return valves. It works in the same way as a piston in that it moves up and down to increase and decrease an enclosed volume, but instead of sealing rings the gap is enlarged and a rubber diaphragm is inserted to give 100% sealing. The diaphragm has a limit to its travel and is also limited in the pressures which can be withstood across it.

The plunger pump works the same as the diaphragm pump but uses seals instead of a diaphragm.

A number of novelty type pumps have been suggested:

- a venturi powered pump uses high pressure air through a venturi to suck in the diaphragm. When fully in, the high pressure air is used to blow the diaphragm out again (Fluid Power, 1974).
- a double disc arrangement has been suggested (Burrage, 1979) by which the discs are able to be deformable and operate 180° out of phase from each other. The first disc seals the outlet off from the inlet and deforms, displacing a certain amount of fluid. The second disc down stream of the first, seals and deforms while the first is being withdrawn, allowing the deformation of the second disc to pump a certain amount of fluid. The discs in effect act as non-return valves when they make the seal and then act as a diaphragm when they deform.

2.5 CURRENT AND PROPOSED PUMPING SYSTEMS

Since the first small water pump powered by the sun was made by Auteuills in 1885, there have been a multitude of small pumps produced or developed to the prototype stage. A large study of the solar water pumps for use in developing countries has been carried out by Halcrow and Partners (1983b). The study generally focused on units with greater than 50 W output. A comparison of all the pumps found in previous literature can be made using Table 4.

By far the majority of small water pumps and power systems (which mention solar water pumping as an application) use sheet and tube collectors to boil the working fluid. The vapour then powers a simple piston or diaphragm expander, but does not maintain a Rankine cycle fully. None of the papers on water pumps uncovered in the literature search reveal any more than prototype production or prototype testing in situ. A similar trend is evident in the field of larger power units (1 to 5 kW) (Halcrow and Partners, 1983b) where the few units which made it past the prototype stage have faltered during field trials.

Of the small systems surveyed, the majority cite developing countries as their probable major market, but very few have carried out even a basic viability analysis for such situations.

Table 4. Summary of Literature

Reference	**Prime mover**	**Pump type**	**Collector type**
Trihey (1976)	bi-metallic	piston	-
Kishore et al. (1986)	special Rankine	"	sheet + tube
Catteneo et al. (1983)	piston expansion	"	"
Imou et al. (1987)	Rankine	turbine	"
Mahendra et al. (1980)	special Rankine	diaphragm	"
Schoeppel (1976)	hydride dehydride	turbine	-
Kelsey (1974)	piston expander	pressure	concentrating trough
Mac Cracken (1959)	piston and U tube	tube diaphragm	-
Mac Cracken (1960)	"	"	-
Speidel (1977)	special Rankine	piston	flat plate
Booth et al. (1967)	special Rankine	bellows	"
Boldt (1977)	"	pressure	flat plate + reflectors
Chadwick (1980)	"	diaphragm	area focus
Umarov et al. (1976)	"	"	hot box
Jenness, (1961)	"	pressure	area focus
Thureau et al. (1976)	"	bellows and diaphragm	sheet + tube
Vinayagalingam (1981)	"	diaphragm	"
Rao et al. (1976)	"	pressure	flat plate
Burton (1983)	"	piston	sheet + tube
Stubbs (1980)	"	diaphragm	"
Bhattacharya (1978)	"	"	flat plate
Zhi-chen (1985)	"	"	sheet + tube

Table 4. Summary of Literature (continued)

Reference	Fluid used	Power output	Efficiency
Trihey (1976)	-	-	-
Kishore et al. (1986)	R11	50 W	0.7 %
Catteneo et al. (1983)	Freon 12	153 W	2.8 %[1]
Imou et al. (1987)	Freon 114	51 W	0.5 %
Mahendra et al. (1980)	Freon 113	3.2 W	0.39 %
Schoeppel (1976)	Hydrogen	-	-
Kelsey (1974)	Freon 112 or 113	-	-
Mac Cracken (1959)	Methyl alcohol	negligible	-
Mac Cracken (1960)	-	-	-
Speidel (1977)	R 11	-	1.5 %
Booth et al. (1967)	-	-	-
Boldt (1977)	water or ethyl alcohol	-	-
Chadwick (1980)	cyclo-pentane	-	-
Umarov et al. (1976)	Freon 113	33 W	-
Jenness, (1961)	Water	13.5 W	4.5 %[1]
Thureau et al. (1976)	-	-	-
Vinayagalingam (1981)	Freon 113	-	1.8 %
Rao et al. (1976)	n-pentane	90 W	4.4 %
Burton (1983)	R 113	8 W	0.21 %
Stubbs (1980)	-	-	-
Bhattacharya (1978)	n-pentane, CS_2 C_3H_6O	21 W	2.4 %[1]
Zhi-chen (1985)	Pentane	5 W	0.05 %

[1] excluding collector

Table 4. Summary of Literature (continued)

Reference	Current status	Collector generates	Positioned up or down well
Trihey (1976)	Patent	-	up
Kishore et al. (1986)	Prototype	vapour	up and down
Catteneo et al. (1983)	Tests in situ	vapour	up and down
Imou et al. (1987)	Test bed	vapour	up
Mahendra et al. (1980)	Prototype	vapour	up
Schoeppel (1976)	Patent	-	-
Kelsey (1974)	Patent	vapour	up
Mac Cracken (1959)	Patent	-	-
Mac Cracken (1960)	Patent	-	-
Speidel (1977)	Hypothetical	vapour	up
Booth et al. (1967)	Patent	vapour	-
Boldt (1977)	Prototype	vapour	up and down
Chadwick (1980)	Patent	vapour	up
Umarov et al. (1976)	Prototype	vapour	up
Jenness, (1961)	Hypothetical	vapour	up
Thureau et al. (1976)	Patent	vapour	up and down
Vinayagalingam (1981)	Hypothetical	liquid	down
Rao et al. (1976)	Prototype	vapour	up and down
Burton (1983)	Prototype	liquid	up
Stubbs (1980)	Patent	vapour	up and down
Bhattacharya (1978)	Prototype	vapour	up and down
Zhi-chen (1985)	Prototype	vapour	up

3 TOWARDS THE FINAL SOLAR WATER PUMP CONFIGURATION

The road towards an appropriate prime mover - pump combination, to fit with the ideals expressed in Chapter 2 involves many compromises. The characteristics of power output of the solar collector and prime mover combination should complement the pump's power demand. A system with well matched components should be achieved while simplicity of construction and ease of maintenance are held paramount.

3.1 THE PRIME MOVER AND PUMP

The matching of prime mover and pump to fulfil operational, construction and maintenance ideals requires a unit specifically designed to meet these criteria. The unit should be of a simple design and construction in order to enable low cost manufacture and semi-skilled maintenance. Ideally such a unit should have no wearing or bearing surfaces. If wearing or bearing surfaces are necessary such surfaces should be easily replaced, with preferably a part which is common in form and tolerances to that used in other similar machines. A moulded plastic part would be ideal, as its tolerances and form would be constant.

The prime movers which use rotating power transmissions would generally be inappropriate due to the manufactured tolerances needed for the rotating members and bearings. The complexity of gearboxes needed to translate the output speed to match the ideal speed of the pump unit would make them also inappropriate.

3.1.1 Operating Features of Previous Pumps

The simplest proposal for an integral prime mover and pump combination is the vapour piston (Rao et al., 1978)(Jenness, 1961)(Rao et al., 1976). The vapour piston uses the high pressure vapour produced in a solar collector to force water out of the cylinder via a non-return valve. The vapour then condenses and sucks water up from the well via a second non-return valve. In these units the valves are the only moving elements (see Fig. 8). The problems with such pumps is the limitations placed on the working fluid. It must not make the water unsuitable for human consumption, and thus a great limitation is placed on the choice of available working fluids. The fluid must also have an appropriate (low) boiling temperature to enable high collector efficiency. Pressure generated during boiling and condensation must be appropriate to pump the water from a depth to the desired height. One large practical problem with this device is that the vapour condenses at the water/vapour interface, but as the water is being expelled, the condensation occurs at both the water/vapour interface and the walls of the cylinder which have been chilled and exposed by the descending water.

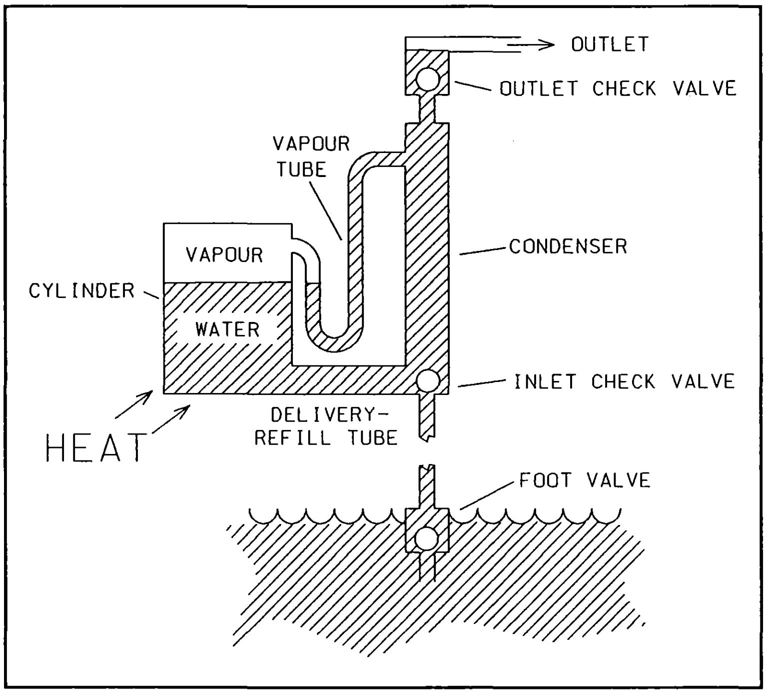

Figure 8. Vapour Pump

The next progression from the liquid piston idea involves the use of an insulated plug to cover the water surface and ideally has a short stroke, large diameter cylinder (Mac Cracken, 1957 and 1959). The use of an insulated plug significantly reduces the undesired condensation on the water/vapour interface (see Fig. 9), but leaves the toxic working fluid and appropriate pressure problems unsolved.

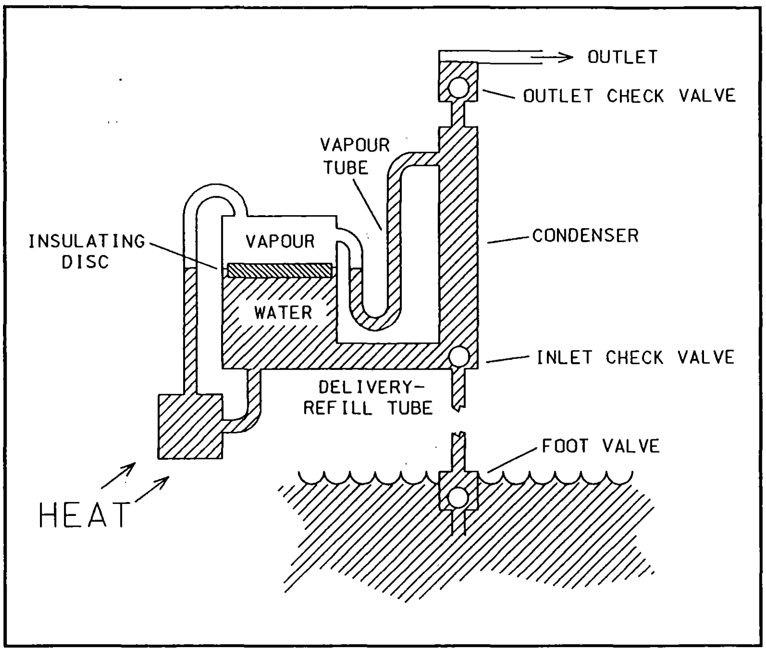

Figure 9. Vapour Pump With Insulating Plug

A toxic fluid can be used by the inclusion of a membrane to separate the water from the vapour. This enables any working fluid to be used provided it is compatible with the diaphragm (the majority of the small solar thermal pumps are of this arrangement (Bhattacharya at al. 1978) (Umarov et al., 1976) (Vinayagalingam, 1981) (Chadwick, 1980)). The remaining problem with the single diaphragm unit, is the possibility of incompatibility of the working fluid pressures with the particular pumping application.

The system as shown in Fig. 10 will deliver water from and to approximately the same pressure as the working fluid on the other side of the diaphragm. For example, if the working fluid pressures are 150 kPa and 50 kPa absolute for the collector and condenser respectively, then the pump will retrieve water at 50 kPa and deliver it to 150 kPa absolute. This equates to taking water from five metres below the pump and delivering it to five metres above the pump unit. Knowing both the depth below and height above the pump to which the water is to be conveyed, the pressure at which it is desired to have the working fluid is known. (Note: it is in practice necessary to have an excess of actuating pressure to accelerate the fluid and overcome pipe frictional and fluid flow losses.)

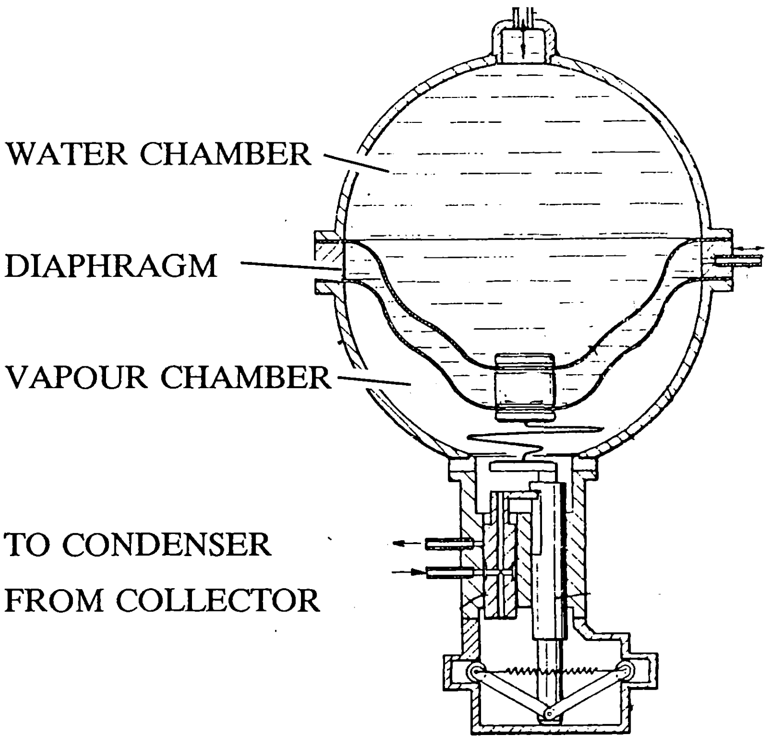

Figure 10. Diaphragm Pump
(from Thureau et al., 1976)

The most efficient point for the collector-prime mover combination is temperature dependent. As the collector deals with a saturated vapour, the pressure and temperature of the working fluid are related. If the Carnot cycle efficiency is multiplied by the collection efficiency, which is also temperature dependent, one gets a graph like Fig. 11. From this graph the optimum temperature/pressure for the working fluid can be found. This optimum working fluid pressure must be sufficient to pump the water to the desired height above the pump. The suction pressure/temperature is dependent on the condenser unit, as the minimum temperature possible is that of the fluid cooling the condenser unit. For air cooled devices this minimum temperature is equivalent to the wet bulb temperature of the air (i.e. temperature to which air falls when it has absorbed moisture up to the 100% humidity point). For water cooled devices using water pumped from the well, the water temperature is the average yearly temperature for that location (only if the depth is greater than two metres (Matthess, 1982)). (If the well is deeper than 15 metres then geothermal heating occurs, if it is less than two metres seasonal variations in water temperature are found, and if in the top several centimetres a daily variation in temperature is apparent.)

The pressure developed in the condenser should be low enough to draw water from the desired well depth. It must also be noted that after a period of regular draw off from the well, the operating depth of the well will fall (Clark, 1970) . The desired heights from which water is drawn and to which it is pumped for design purposes of this pump are 6.5 metres and 1.87 metres respectively.

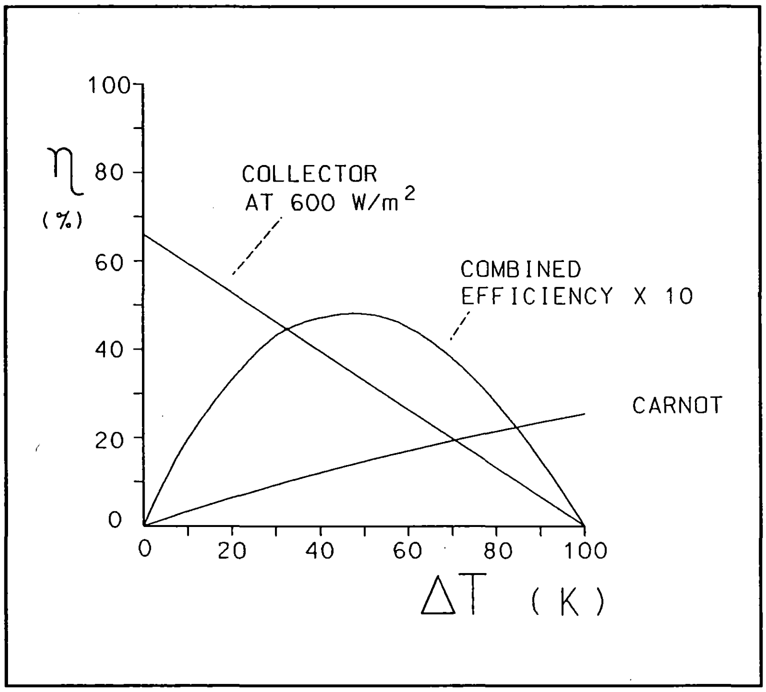

Figure 11. Combined Efficiency (T_c = 290 K)

It is possible to run a pump with a working fluid which does not produce one of the desired pressures but produces an excess of the other. This excess pressure can be stored for use during the second part of the cycle by use of a spring or a dead weight acting upon the diaphragm. This has been done by Zhi-Chen et al. (1985), Sharma et al. (1980) and Thureau et al. (1976). The main problems with such systems are as follows:

A) For a moderate depth well, the majority of the work is done by the low pressure stage of the working fluid's cycle. At this low pressure any leaks in the system will tend to draw air into the system. This unwanted air will give a residual background partial pressure, which being relatively independent of temperature, will hinder the low pressure stage of the cycle. If too much air leaks into the system the lowest pressure achievable by the condenser will not be sufficient to draw water up from the well.

B) The use of a storage spring leads to large stresses on the diaphragm by having a differential pressure acting over the diaphragm's surface area. This pressure imposes a restrictive limit to the length of travel of the centre of the diaphragm. The size of spring can also be prohibitive for larger pressure storage (e.g. for a 0.1 m^2 diaphragm area, to store an excess pressure of 100 kPa, a 1 tonne spring force is needed; this is the size of an automobile spring).

It is possible to run a pump with a working fluid which does not produce one of the desired pressures but produces an excess of the other by using two diaphragms working against each other e.g. see fig 12. Such systems are common in pneumatic pumping applications, but this unit has almost twice the componentry needed, i.e. two diaphragms, two vapour chambers, two water chambers and two pair of non return valves. The duplication could be avoided and will enable the capital cost of the pump to be reduced.

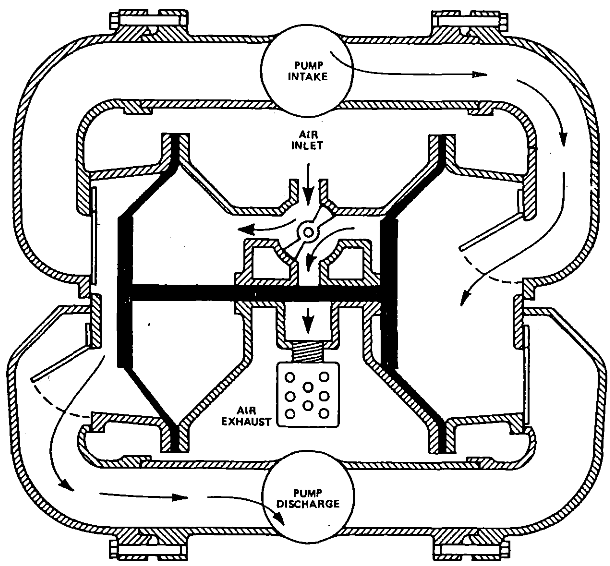

Figure 12. Opposing Diaphragm Pump (from Karrasik et al., 1984)

3.1.2 Operating Features of Final Pump

In order to overcome the problems of system leakage at low pressure, and spring size to get a fluid with appropriate pressures, the operation of the water side can be inverted. This inversion enables the majority of the work to be done by the pressure generated in the solar collector and a small amount of work to be done by the suction produced by the condenser, this is approximately proportional to the energy available from the working fluid at each of these stages. The pump now works as in Fig. 13. To expel the water, the diaphragm is pulled down by the low pressure developed in the condenser (Fig. 13 left). To induce water, the diaphragm is pushed up by the working fluid's pressure (Fig. 13 right). So an increase in volume of the vapour chamber increases the volume of the water chamber.

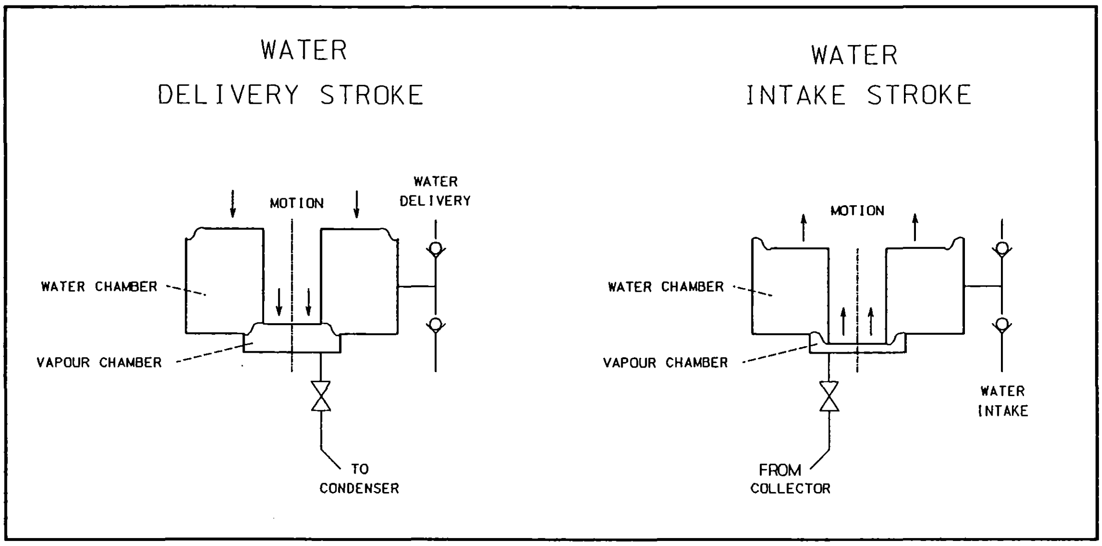

Figure 13. Double Diaphragm Operating Principle

This system is most appropriate because where the bulk of the work is done, a system using a low boiling point working fluid is at its most able to do work and has thus replaced the large low pressure process with a large high pressure process. The small high pressure process has been replaced by a small low pressure process. This system has one major benefit and one major drawback. The drawback is with the diaphragms, which now must withstand large pressure differences between their two sides. The pressure difference causes a stress which limits the width of the diaphragm and thus the stroke of the pump's central unit. To enable operation under such conditions a large aspect ratio is desirable, i.e. the pump body is of large diameter but has a short stroke. The major benefit is

that the ideal working pressure/temperature for the collector-prime mover combination can be achieved irrespective of pumping heads, by varying the ratio of outer diameter of the water chamber to that of the vapour chamber.
Fig. 14 demonstrates how the diameter ratio (ratio of water chamber diaphragm's mean diameter to vapour chamber diaphragm's mean diameter) can be altered so that the vapour will operate at its optimum temperature/pressure despite the different pumping heads. It is possible to increase or decrease the diameter ratio and thus achieve a pressure balance for the collector-prime mover-water pumping system. An example of such a pressure balance over the whole double diaphragm pump is given in Appendix 3.

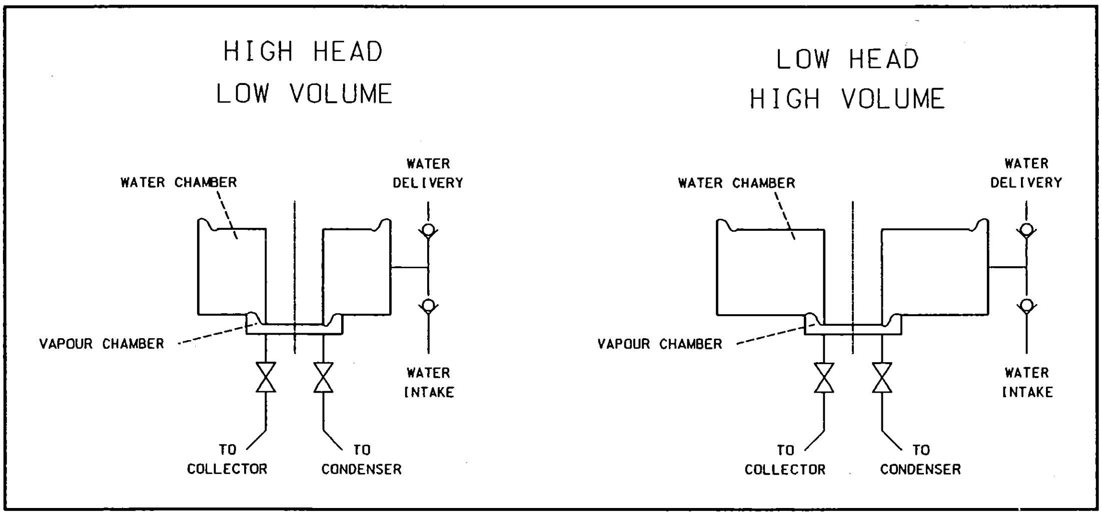

Figure 14. Diameter Change to Accommodate Water Head

The combined water chamber and prime mover will not however, operate at 100% efficiency. The prime mover is limited by its thermodynamic conscience to the Carnot efficiency. If the prime mover does not follow an ideal cycle, its efficiency will be lower still. Further losses will occur due to condensation of the working fluid onto the walls of the vapour chamber. Inevitable dead volume in the vapour chamber will lead to losses when this volume undergoes pressurisation and depressurisation without additional work being performed. The water chamber will also have its inefficiencies, in that it needs to accelerate the water into and out of the chamber; this work is lost, but is nonetheless essential. There will also be losses due to the motion of the water in the pipes and fittings. A trade off must be made in terms of the cycle time and rate of water acceleration into and out of the water chamber.

3.2 VALVING TO CONTROL THE PRIME MOVER/PUMP UNIT

A prime mover-pump unit (whose operating principles have been described in the previous section of this chapter) needs a valve or controlling system to control the cycle, i.e. allow entry of the vapour from the collector and exhaust to the condenser at the appropriate times in the pumping cycle. There were two main options for operation of the cycle, each had its benefits and shortcomings.

3.2.1 The First Option - Hot Plate Pump

This first operating system was worked through with three different valve set-ups. The first operating system's pump was assumed to have a solar heated base, onto which liquid was poured, the liquid vaporised forcing up the diaphragm. The valve then opened and the condenser reduced the pressure and pulled the diaphragm back down. So the valve performed two functions, measuring out the liquid and then exhausting the vapour to the condenser.

3.2.1.1 Mercury valve

The first valve set-up considered was not very realistic in its practicality as it utilised a liquid valve. The liquid valve was given consideration due to its having no wearing or contacting parts involved in the valve at all. The valve ideally utilised a very dense fluid (probably liquid Hg) which was forced through tubes of different cross sections. In transferal between the tubes the Hg measured out the volume of working liquid to be injected and controlled the opening of the vapour chamber to the condenser. The working principle assumed that when the diaphragm hit the top of its travel the pressure would continue to rise because more working liquid was injected onto the hot surface of the vapour chamber than was needed to expand the vapour chamber. When the pressure rose above a certain threshold, where the mercury in U tube 1 reaches the larger chamber (Fig. 15b), an increase in pressure would see all of the mercury flow around the bottom of the U tube and decrease in height until all the Hg is in the large chamber. The vapour then experiences very little hindrance in flowing through the mercury and into the condenser (Fig. 15c). When the pressures equalise in the condenser and vapour chamber, a column of Hg will flow from the large chamber back down into the U tube to start the process once again (Fig. 15a).

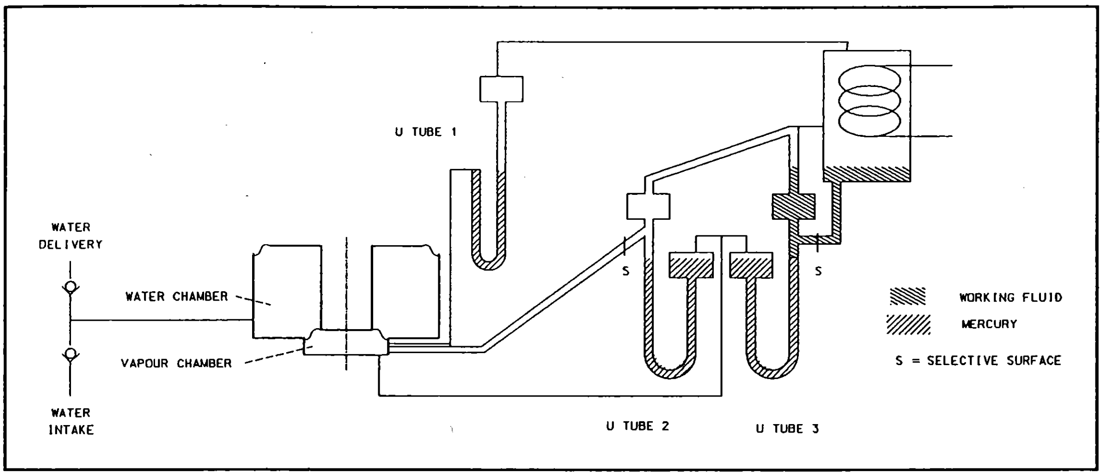

Figure 15a. Liquid Valve

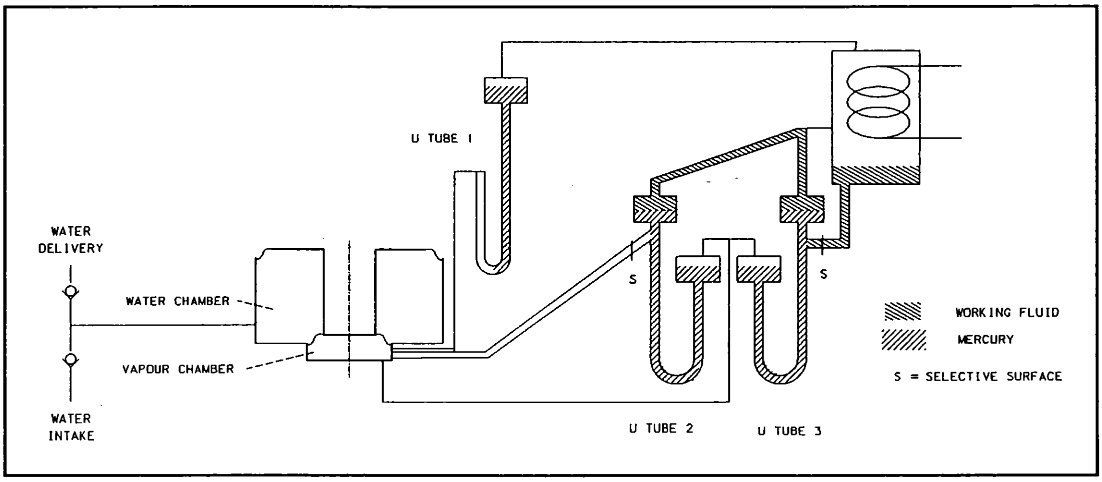

Figure 15b. Liquid Valve

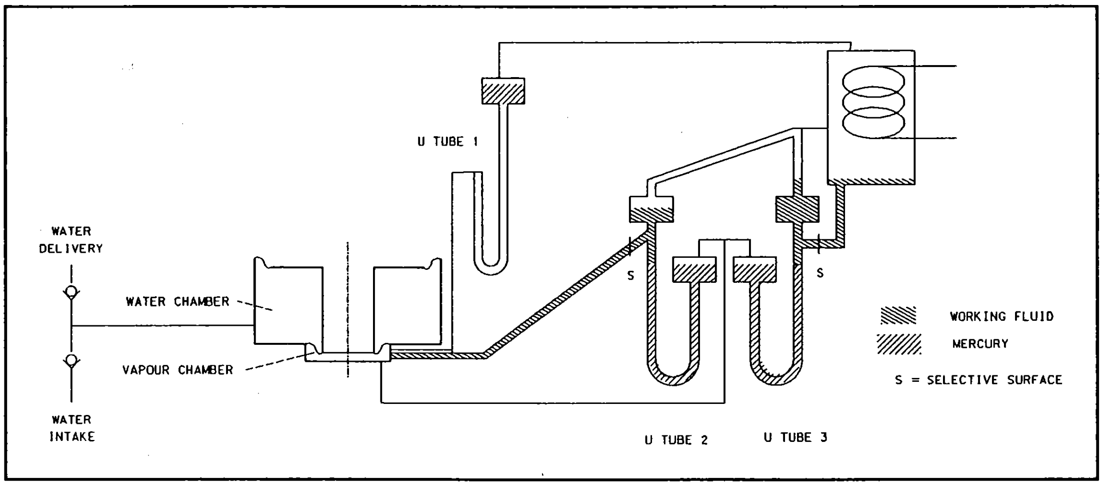

Figure 15c. Liquid Valve

The liquid measuring system for this valve set-up was complicated and involved the use of a selective filter which would allow the passage of working liquid, but not the dense mercury (this is possible because of some surfaces distinguishing between the mercury which is non-wetting and the hydrocarbon working liquid which is wetting).

After the injection of the working liquid into the vapour chamber, it is vaporised on the heated base of the chamber, increasing in pressure. The two measuring U tubes (tubes 2 + 3) start with their levels on both sides at an equal height (Fig. 15a). In U tube 3 the right chamber is full of working liquid above the mercury. As the pressure increases, this working liquid is forced upwards and starts pouring down the inclined tube to rest above the mercury of U tube 2. As the pressure continues to rise the mercury rises until the vapour chamber is opened to the condenser. The working liquid is measured out in the top section of the inclined tube joining U tubes 2 and 3, from which any excess is lost through the top breather tube back to the condenser/liquid reservoir (Fig. 15b). As the vapour pressure falls due to the condenser being opened to the vapour chamber, the mercury levels drop, leaving a measured quantity of working liquid in U tube 2. As the level drops further the mercury passes the selective membrane and the measured quantity of working liquid flows through a non-return valve into the heated vapour chamber to restart the process (Fig. 15c).

This valving idea had likely problems with:

- flow rates through the selective membranes
- the operation of the opening of the vapour chamber to the condenser assumes that the pressure during motion of the diaphragm will not exceed that when it is at the top of its travel. Such a pressure regime is possible but is inefficient if too high a pressure is set to be the change-over pressure for the operation of U tube 1.
- if the water pipes were to become constricted, it would be possible for the pump to cycle without doing any work.
- the cost of mercury could be prohibitive.
- the unit would need precise volumes of mercury in the tubes to work well.

- the tubes would be exposed to external forces and thus vulnerable to interference.
- the unit is not multi-purpose because the tube lengths and Hg volumes would have to be altered for different pumping heads.
- mercury is a poisonous substance.

3.2.1.2 Double diaphragm valve

The second valve considered operated as follows: A measured quantity of working liquid vaporises in the heated vapour chamber. The increase in pressure forces the diaphragm up until either there is no motion of the diaphragm or the diaphragm reaches the top of its travel and ceases motion. This cessation of motion is the signal for the valve to open and admit the vapour to the condenser. The condensation of the vapour lowers its pressure and the diaphragm is sucked back down. On reaching the bottom of the chamber the diaphragm pushes the valve shut, closing off the condenser and simultaneously injecting the next charge of working liquid into the heated vapour chamber.

This valve takes the form of a double diaphragm unit which uses certain pressures within the pump to open the valve and to hold it closed, Fig. 16. The three chambers of the valve were open to different parts of the pump:

- the bottom chamber was open to the condenser's liquid side and vapour chamber via two non-return valves,
- the middle chamber was open to the pump's water chamber,
- the valve's top chamber was always open to the condenser's vapour side and occasionally the vapour chamber via the valve head.

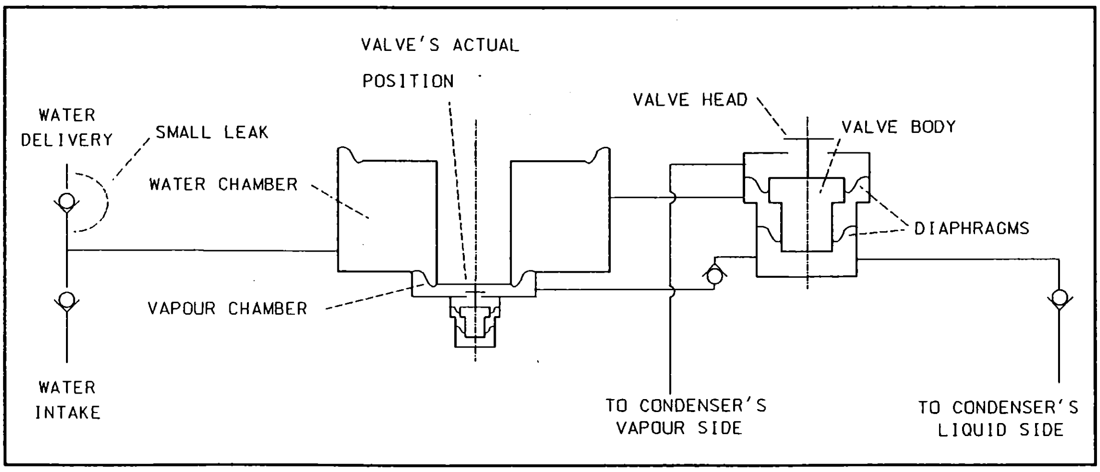

Figure 16. Double Diaphragm Valve

For the purpose of the valve's operation, a small leak was maintained past the top water non-return valve, ensuring that when neither of the water valves were open, the water chamber would be at the pump's water delivery head pressure.

The valve operated as follows (see Fig. 17): the injected liquid vaporises and the diaphragm starts to move upwards, the water chamber takes in water and the valve is held shut by the pressure of the vapour chamber acting on the valve head (Fig. 17a left). On reaching the top of its travel or ceasing motion the pressure in the valve's central chamber increases due to it being attached to the water chamber. The water chamber increases in pressure due to the small, controlled leak to the pump's delivery head. Because the diaphragm of the valve's top chamber is larger than that of the bottom chamber, the force balance within the valve changes so the valve head opens (Fig. 17a right)(the delay for this opening to occur can be controlled by the size of tube attaching the valve to the water chamber, i.e. the smaller the tube, the longer it would take to fill and thus compress the middle chamber). On opening the vapour chamber to the condenser, the pressure in the vapour chamber falls. The valve continues to be held open by the same forces which forced it to open (Fig. 17b left). Just before the diaphragm reaches the bottom of its travel however, the valve head's spring, impacts with the top of the vapour chamber and forces the valve shut as the diaphragm reaches the bottom of its travel (Fig. 17b right). The bottom chamber of the valve is compressed in this process, injecting some working liquid through a non-return valve into the heated vapour chamber. This injected liquid boils, increasing in pressure until the diaphragm starts rising. Until the diaphragm starts rising the pump's water chamber remains at the higher pumping pressure due to the leak. The initial travel of the diaphragm opens the low pressure water intake to suck in water. This low pressure is also in the valve's central chamber and it holds the valve head shut (Fig. 17a left).

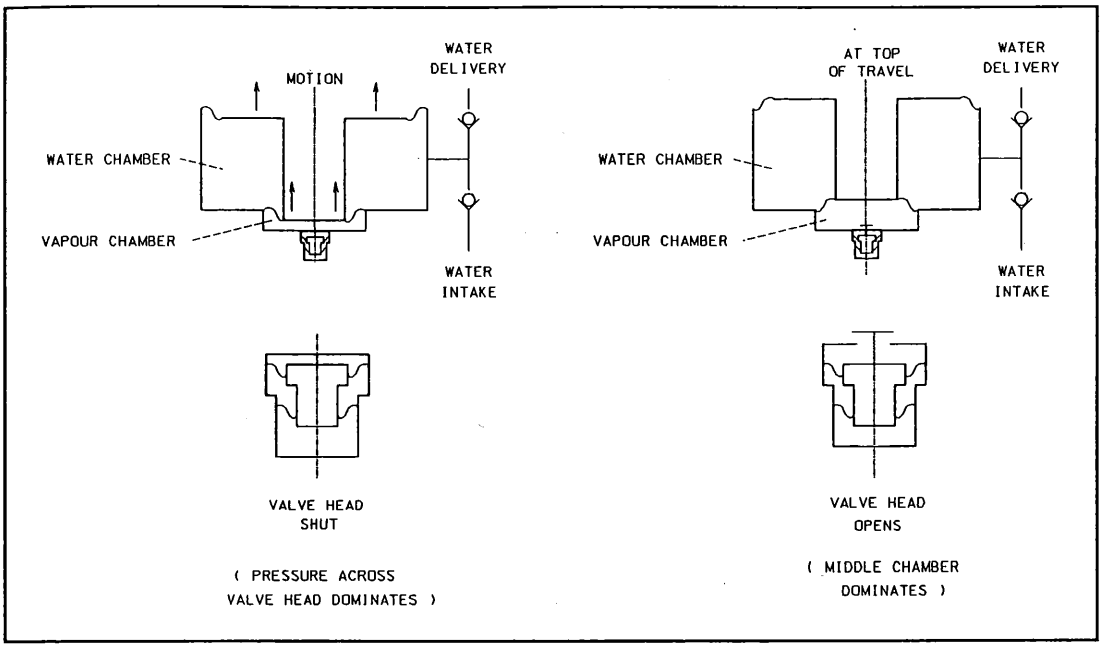

Figure 17a. Double Diaphragm Valve's Operation

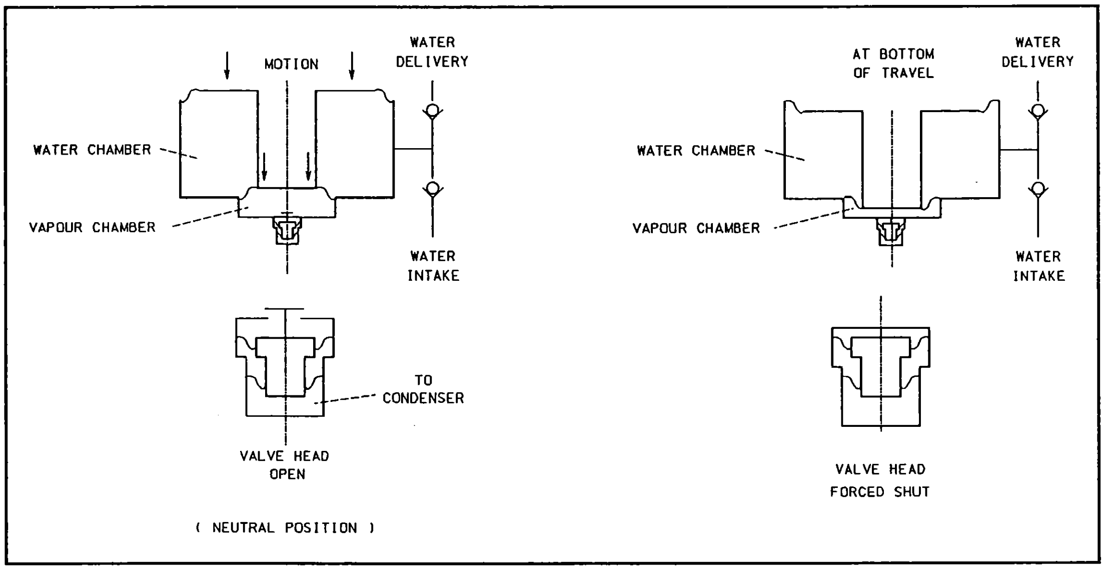

Figure 17b. Double Diaphragm Valve's Operation

The positive features of this system are:

- the collector and pump units are separable while being fully self-contained and thus are easily and readily assembled for immediate operation without purging,
- the introduction of only a certain measured amount of working liquid ensures good use is made of all the vapour, i.e. no excess vapour pressure is produced,

- the change to condensing is automatic when the diaphragm motion is stopped, ensuring the diaphragm returns to its bottom position and a new charge of working liquid is introduced for the next cycle, irrespective of the power being provided by the collector. This feature ensures that the pump is always ready to start another cycle,
- the boiling will quicken to take into account a greater heat input into the pump so it will, to an extent, match the power being provided by the solar collector.

Tests were carried out on a prototype of this valve with the pressures in the valve's chambers being simulated. It performed as expected, except for an undesirable extra squirt of working liquid when the central chamber's diaphragm ballooned into the bottom chamber because of the pressure in the central chamber. The main disadvantage with this system was the fact that the valve would need extensive setting up to get it behaving fully as desired. To change pumping heads the valve would need extensive setting up once again. This setting up would involve changing the valve diaphragm and valve head areas. The size of the water leak across the water valve and size of the tubes going to the valve from the water chamber would also need to be altered.

Work on both the first valve concept and the second valve prototype was discontinued for similar reasons:

- neither were multi purpose because the variation in water table level would cause the first to cycle without pumping and the second to pump only a minimal amount of water.
- to allow for different pumping heads the tube lengths of the first valve would need altering and the second valve would need to have the relative areas of diaphragm in the valve changed along with the size of the pipe joining the valve's water chamber to the pump's water chamber.

To overcome some of these problems it was decided to operate a valve directly from the position of the diaphragm.

3.2.1.3 Mechanically indexing valve

The third valve considered, worked as follows: when the diaphragm reaches the top of its travel the valve opens the vapour chamber to the condenser. The valve stays open until the diaphragm reaches the bottom of its travel, at which point the condenser is shut off and a new charge of liquid is poured into the heated vapour chamber. This valve is operated by a lever with 2 springs attached, to form an over centre device, see Fig. 18. This ensures the valve is fully in either of its two operating positions. The valve consists of a cylinder with two radial holes drilled in it; one hole being significantly larger than the other. In its first position the valve has the small hole opening the way for the high pressure vapour to go to the condenser. At the same time the larger hole is being filled with liquid from the liquid side of the condenser. When the diaphragm hits the bottom of its travel the valve toggles over to its second position. The larger hole (which is no longer open to the condenser's vapour side) pours the liquid into the heated vapour chamber. When the diaphragm reaches the top of its travel the valve toggles over again to restart the process.

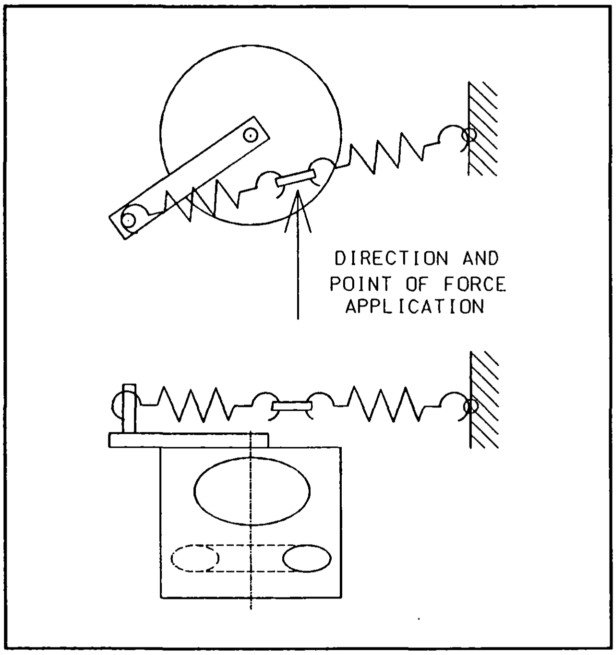

Figure 18. Mechanically Operated Valve

This concept was disregarded because:

- it had bearing elements which were also required to develop a pressure seal and thus needed fine tolerances.

- the valve is awkward to fit into the vapour chamber without creating excessive dead volume.
- the positions where toggling would occur would depend on the pressure acting on the sealing surfaces of the valve. This would lead to inaccurate and inconsistent operation.
- the amount of liquid injected into the heated vapour chamber must take into account all possibilities, ensuring the diaphragm will always reach the top of its travel so another cycle will take place. This would be inefficient because an excess of liquid would always have to be boiled.

3.2.2 The Second Option - Vapour Pump

A major physical constraint of the pump set-up above was the reliance on the hot plate to boil the liquid, using the energy supplied by the solar collector. This set-up needs two large heat transfers to take place; firstly from the collector's fluid to the bottom of the pump's vapour chamber and then from the bottom of the pump's vapour chamber to the boiling liquid. In order to boil the liquid in the required time, a temperature difference between the plate and the fluid's boiling point of 15K was needed. The 15° temperature loss at the vapour chamber interface was thought to be excessive. The heat transfer from the collector to the vapour chamber could take place using a heat pipe type system, where the liquid is boiled in the collector and condenses on the vapour chamber's outer surface. The heat pipe system would have only a couple of degrees temperature loss associated with it. To avoid the 15° temperature loss a system using vapour at high pressure directly from the collector to power the pump was investigated. The final valve for the prototype pump was inspired by Stubbs (1977) and Zhi-Chen et al. (1985). The valving principle was for the valve to simultaneously open the vapour chamber to the solar collector and shut off the condenser. In the other position of use, the valve would simultaneously shut off the condenser and open the vapour chamber to the solar collector. The pump and valve operate as follows: the two way valve which toggles between its two operating states, links the vapour chamber directly to either the solar collector or the condenser. The valve's change from one position to the other is controlled directly by the pump's displacement. Initially the displacement of the lower diaphragm compresses a valve spring. Before the spring is fully compressed however, a stop is reached.

The stop causes the valve to be unseated from its stable position, where it was being held by the differential pressure across the seated portion of the valve. The valve is then delivered to its opposite position by the spring. Once in the new position, the valve is held in place by the spring until the pressure in the vapour chamber has changed sufficiently for the valve to be held in place by the differential pressure. As the diaphragm returns towards its initial position, a similar spring and stop action toggles the valve back to its initial position. This in effect makes the valve end-biased according to the pump diaphragm's displacement.

3.3 CONDENSER

The condenser has the purpose of condensing the working fluid vapour which was generated by the solar collector and has had power extracted from it via the pump-prime mover unit. By condensing the vapour at a lower temperature than that at which it was created, the pressure in the condenser is lower than that of the collector. It should be noted that in general, the vapour pressure does not decrease linearly with condenser temperature but on a logarithmic scale, so a reduction in temperature further from the collector's temperature produces an increasingly disproportionate reduction in vapour pressure of the working fluid. A range of possible condensers was worked through, given two different siting positions and several possible different condenser configuration options.

There is the possibility of having the condenser external to the pump-prime mover unit as this would give the choice of different means for cooling the condenser. The choices of cooling fluid are: the pumped water and the air at ambient temperature, or both in the guise of an evaporative cooling unit. As is mentioned in Chapter 2, the ground water temperature is at the average temperature for that location and is in summer, significantly cooler than the ambient air temperature. In the cooler seasons the water temperature may be higher than ambient air temperature but there is usually less demand for water in these seasons.

For reasons of heat transfer, the water cooled condenser is less expensive. The basis for this is the greater specific heat of the water, enabling the water cooled condenser to be significantly smaller than its air cooled counterpart and therefore less expensive.

The third possibility is an evaporative cooling unit which can bring the condenser temperature down below ambient dry bulb temperature. This unit would work by having the condenser's external surface covered with a thin layer of water. This helps cooling because along with the conductive, convective and radiative heat transfer there is also mass transfer to the air. This mass is made up of the water which evaporates and takes with it the latent heat of vaporisation and provides significant cooling. The mass transfer effect is less significant the further the condenser temperature is from the water's boiling point. In order to work properly the vaporising liquid must be in a thin surface film over the condenser surface. The most practical means of achieving the thin liquid film is to use a wicked surface or a surface coating which the water wets, thus ensuring a thin, even coating.

The option of having the condenser in the pump unit is beneficial from the point of view that it is desirable to have the unit as small and use as few components as possible, i.e. the pump body also serves the function of being the condenser housing.

The final condenser unit in the prototype takes the form of a spiralled coil of copper tube sitting in the water chamber of the pump unit. The major drawbacks of this placement are the large circular structure (pump casing) which has to be manufactured and the added dead volume inside the water chamber. This dead volume is of no consequence when pumping but is detrimental to start-up with no or low water level in the pump. Excess dead volume requires that the pump be primed with water to immerse the coil, enabling operation. Also, the lower the dead volume of the water chamber, the more likely it is that the unit could be self priming and will reduce the number of strokes for this priming to take place.

3.4 SOLAR COLLECTOR-FLAT PLATE STRUCTURE AND THEORY

The flat plate collector comes in an assortment of configurations, each with different ideals and applications. The typical flat plate collector is a sheet and tube collector, Fig. 19. Such collectors use a sheet with a blackened surface to maximise the absorption of radiation. The sheet acts as a fin for the tube which carries the fluid to be heated. There are several major losses in the standard collector:

- conductive heat losses through the bottom and sides.
- convective-conductive heat losses through the top transparent surface.
- radiative energy losses through the top transparent surface.

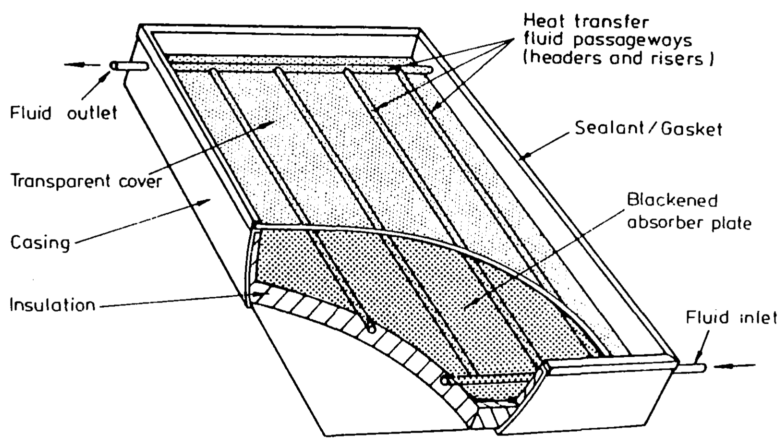

Figure 19. A Typical Solar Collector (from Gillet et al., 1985)

The radiant energy loss is proportional to the sheet absolute temperature to the fourth power minus the ambient absolute temperature to the fourth power. The conductive and convective losses are proportional to the difference between sheet and ambient temperatures. The efficiency of a collector in steady state conditions is most commonly expressed as:

$$\eta = A - \frac{B\,(T_H - T_C)}{I} \tag{3.1}$$

This form of presentation is most convenient as it takes into account both the temperatures and solar radiation intensity, enabling an easily decipherable result for any possible condition of temperatures and radiation. The effect of optical and thermal losses in the collector system (Fig. 20) can also be seen.

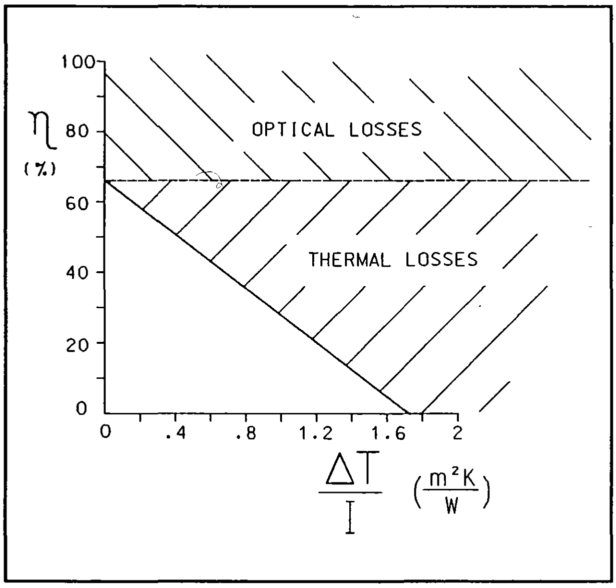

Figure 20. Collector Losses

There have been many different modifications to collectors to reduce both the optical and thermal losses. Selective coatings take advantage of the radiation absorbing properties of glass and some plastics. The incoming radiation peaks in intensity at around the visual range and is close to Planckian at 5800 K, Fig. 21. Comparatively, the radiation put out by a collector surface at 350 K has its peak much lower than the visual range. Glass however, is transparent only to the incoming solar radiation (see Fig. 21). This benefits the collector by having a lower rate of heat loss because most of the radiation emitted by the low temperature collector surface will not be transmitted outwards by the glazing. Enhanced performance can also be obtained by the use of a selective coating on the collector surface. Such surfaces have the characteristic that although their absorptivity, α, and their emissivity, ξ, are equal at any wavelength (as required by Kirchhoff's Law), α (and therefore ξ) is wavelength-dependent exhibiting high absorptivity to incident solar radiation, but low emissivity for the long wavelength radiation characteristic of the collector's surface temperature.

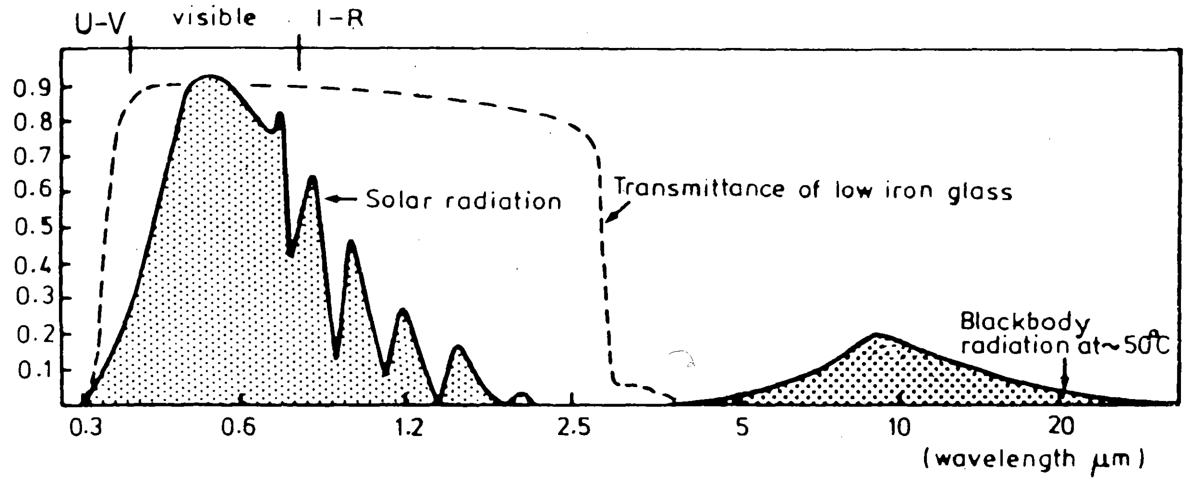

Figure 21. Solar Radiation and the Properties of Glass
(from Gillet at al., 1985)

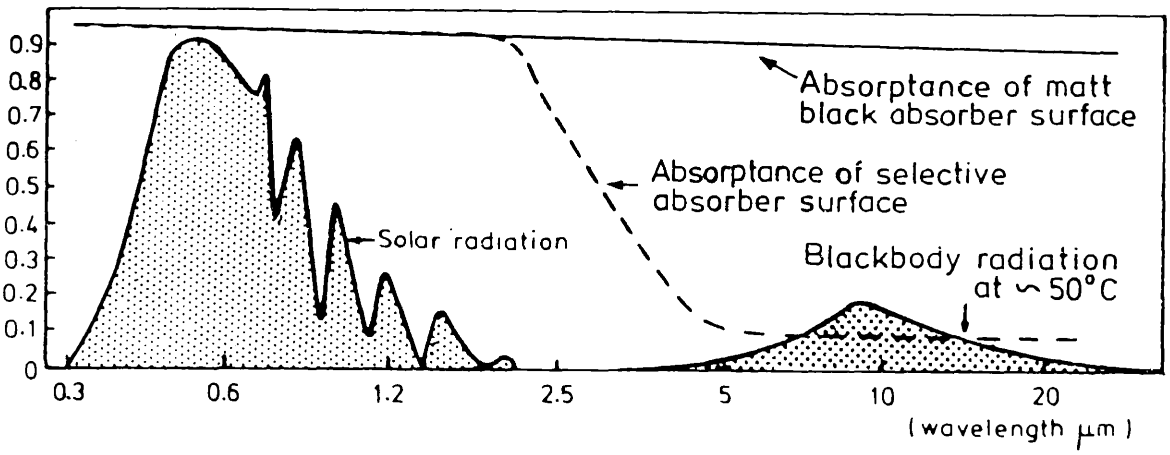

Figure 22. Selective Surface's Characteristics
(from Gillet et al., 1985)

A second method to reduce the heat loss through the top is to double glaze. This provides for a less efficient optical collector because the glass and plastics absorb and reflect between 5 and 15% depending on the type of incoming radiation (up to 100% depending on angle of incidence). It has the benefits however, of providing a double baffle to convective-conductive losses.

Another method of increasing the efficiency of the collector is to reduce the collector plate's temperature for the same fluid outlet temperature, i.e. the fin efficiency of the sheet and tube is increased. This increase in the efficiency for the same outlet fluid temperature means that a lower temperature sheet is there to drive the convective-conductive heat losses and therefore a small amount of thermal radiation and conduction - convection loss is averted.

The increase in fin efficiency is achieved by joining the tube and sheet in different configurations by either spacing the tubes closer together or using different sheet and tube materials. For example, plastic collectors have the tubes without any sheets between them due to the plastic's low thermal conductivity, whereas aluminium sheet and tube has 100 mm of sheet between adjacent tubes.

A more novel way to overcome the fin efficiency was suggested by Meas et al. (1982). Meas used a black fluid (used oil) which was heated directly by absorbing the radiation.

These methods reduce the problem of temperature increase between adjacent tubes, but in a system where cold fluid enters at the bottom and progressively gets heated as it travels to the top, there is a significant temperature gradient with height within the collector. To overcome this vertical gradient the use of heat pipes as a heat transfer media enables the elimination of most of the vertical temperature gradient. The heat-pipe idea can be extended in two ways, either to use the whole collector as a heat pipe ("heat sheet") (Pitkin, 1985) or to boil off vapour in the sheet and tube collector and use the vapour instead of heated liquid (essentially an over-filled heat pipe). The heat sheet has, beneficially a much lower thermal mass. The two phase sheet and tube collector has been tested by Kishore et al. (1984) and is more efficient than its single phase sheet and tube counterpart.

The selection of a solar collector to suit most applications is straightforward, the characteristics are well studied and reported in numerous books and periodicals. For this study a two phase sheet and tube collector is needed to provide the high pressure vapour for the prime mover. The solar collector design was not considered to be a part of this project, and for this reason a standard collector already in use at the University of Canterbury was chosen. It consists of a copper sheet soldered to copper tubes and has one layer of glazing. The main foreseeable problem with the collector is the possibility of the vapour, in travelling from the bottom of the riser, carrying the liquid with it due to the tubes small bore. On further theoretical investigation this was found to be of little concern, as the flow regime of this proportion of vapour in such circumstances will be intermittent slug flow, the characteristics of which are not detrimental to the performance of the two phase collector.

4 DESIGN FEATURES AND EXERGY ANALYSIS

The solar water pump made for simple manufacture, low cost and easy maintenance can be analysed by two methods, in order to find both its efficiency and the operations within its operating cycle that can be most readily improved. Given the basic cycle, both energy and exergy analyses are used to reveal the pump's operating efficiency.

4.1 BASIC CYCLE

The prime-mover/pump unit which resulted from the previous chapter's considerations, operates on a vapour cycle. The cycle is not substantially dissimilar from an organic Rankine cycle. The processes undergone are represented in Fig. 23:

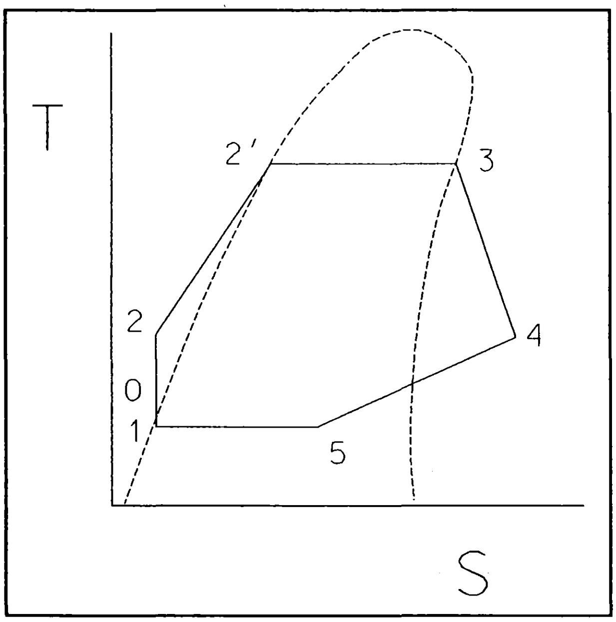

Figure 23. Ideal Pump's Cycle

1 - 2 pressurisation of condensate from a low to high pressure

2 - 2' heating of condensate to saturated vapour line

2' - 3 boiling off of vapour while increasing the vapour chamber's volume

3 - 4 unresisted expansion of the vapour

4 - 5 condensation of the vapour at constant volume until the lowest pressure is achieved

5 - 1 condensation of the remaining vapour while decreasing the volume of the vapour chamber.

Under ideal situations:

1 - 2 is a constant entropy process (isentropic)

2 - 2' is constant pressure (isobaric)

2' - 3 is a constant temperature and pressure boiling of vapour and increase in system volume (isothermal)

3 - 4 is an adiabatic unresisted expansion

4 - 5 is constant volume condensation (isochoric)

5 - 1 is a constant temperature and pressure decrease in volume (isothermal).

P maximum represents the working fluid pressure needed to give a zero net force on the pump when expelling water. P minimum is the working fluid pressure needed to give a zero net force on the pump when taking in water. V_4 is the combination of the vapour chamber's maximum volume and the condenser's volume.

The reasons for selection of n-pentane as the working fluid are presented in Appendix 6 and the state of the pentane at different stages in the cycle is presented in Table 5.

The cycle as described above, using n-pentane as the working fluid, has work performed on the system between states 1 and 2 to pressurise the fluid but it is so small as to be negligible. Energy is added between states 2 and 3 (Q = 451 kJ/kg), performing work (W = 26.9 kJ/kg). From states 3 to 5 no work is done but energy is lost in the condensation process from 4 to 5 (Q = -243 kJ/kg). From states 5 to 1 work is performed by the atmosphere upon the system (useful work W = 8.9 kJ/kg) and energy is extracted in the completion of the condensation (Q = -208 kJ/kg). From the above figures the efficiency of the system is 7.9% which is 53 % of the Carnot efficiency for a system operating between the same temperature limits.

For comparisons with an ideal cycle the prime mover's efficiency will be compared to the efficiency of a Carnot cycle operating between the same upper and lower temperature limits. This policy will give pessimistic values to the efficiency comparisons of the prime mover described above whose real ideal cycle efficiency is that of a Rankine cycle. The use of the Carnot cycle as the base cycle for efficiency relationships was chosen to give a standardised basis for comparison of all prime mover systems throughout the thesis.

Table 5. Working Fluid's States in the Cycle

State	Pressure kPa	Volume m^3/kg	Enthalpy kJ/kg	Entropy kJ/kg K	Temp °C	State
1	50	1.67×10^{-3}	-1640	3.6316	17	liquid
0	101.3	1.67×10^{-3}	-1640	3.6316	17	liquid
2	270	1.67×10^{-3}	-1640	3.6316	17	liquid
2'	270	1.67×10^{-3}	-1519	3.9737	68	sat. liq.
3	270	0.134	-1189	4.9380	68	sat. vap.
4	119	0.312	-1189	5.0618	68	vapour
5	50	-	-1432	4.3543	17	both

One stroke of the water chamber ejects 3 litres of water (ideally) and pumps from 6.5 metres below the pump to 1.87 metres above the pump. It has a 10 second cycle time and has a 1.63 diameter ratio. The maximum and minimum pentane pressures in the cycle are 270 kPa and 50 kPa absolute.

The real process for such a pump would differ in several respects from the ideal. An actual pressure balance on the water pump would reveal a positive resultant force (indicating an excess working fluid pressure) needed to accelerate the water into and out of the water pump. This excess pressure also overcomes losses in the pipe-work and fittings due to fluid motion. The cycle would not necessarily pass through point 4 as the fluid will start condensing on the condenser walls between states 3 and 4, so the end diagram (Fig. 24) will be more slender than the ideal. The pressurisation of the condensate would not be isentropic because extra work has to be done on the working fluid in this stage in order to move the fluid through a length of pipe and two non-return valves.

The efficiency of the unit compared to that of a Carnot cycle over the whole cycle will indicate how effective the whole cycle is, compared with how effective it could be. But to analyse which processes within the cycle have the largest irreversibilities and contribute most to the overall losses, an exergy analysis is of great value.

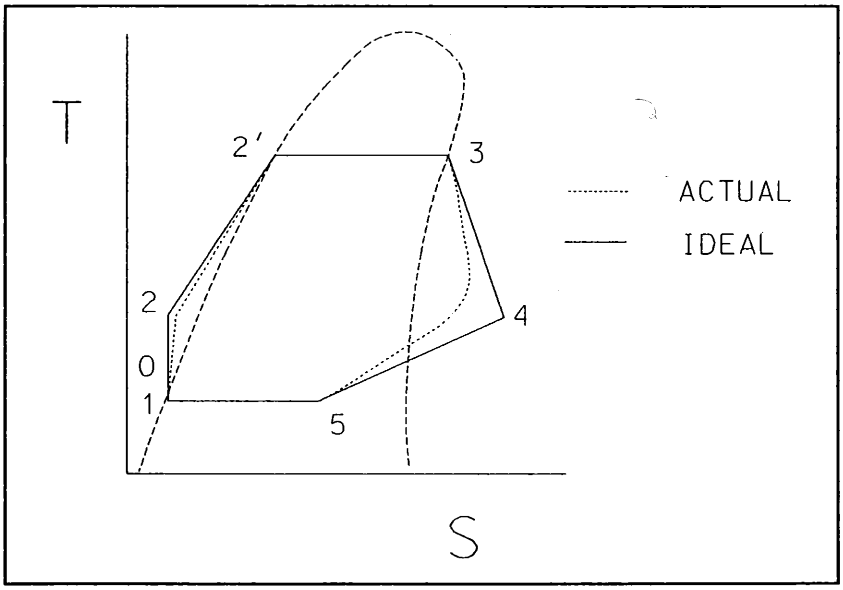

Figure 24. Real Pump's Cycle

4.2 THEORY OF EXERGY ANALYSES

Exergy is a term used to define the maximum amount of work that can be extracted from a medium in going to an arbitrary reference or base state. An energy balance can be performed easily within a system, ie. from the net work output and the energy input the efficiency of the system is known. This analysis has not however, taken into account the maximum actual work that could have been performed. This is the realm of the Second Law of Thermodynamics. Exergy is a general engineering method of applying the Second Law, providing information on how to use energy more wisely and how to extract the maximum work from the source of available energy.

The exergy is always related to a certain base or ground state. In general the base or ground state is chosen to be the lowest temperature of the system or a state at which the system is in equilibrium with its environment. For the water pump, which is influenced by atmospheric pressure, yet is cooled by water which can be cooler than the atmospheric temperature, the base state for the liquid n-pentane was deemed to be at the condenser temperature and at atmospheric pressure.

It is generally possible, from a base condition, to obtain the exergy at any point. Exergy values can be determined by the knowledge of the enthalpy and entropy at both that point and the reference (base) state, along with the base state's temperature. The exergy for a control volume situation being given by ϕ where:

$$\phi = (h_1 - h_0) - T_0 (s_1 - s_0) \qquad 4.1$$

where:

T = temperature, K
h = enthalpy, kJ/kg
s = entropy, kJ/kg K
$_1$ indicates current position
$_0$ indicates base condition

When moving from one point to another, the reference condition cancels out, leaving the difference in exergy to be the same irrespective of the base condition.

Applying the exergy analysis to the water pump was made more difficult because of the constant temperature and pressure condensation and evaporation processes not being able to be expressed as a flow process. Therefore a control mass system had to be adopted.

The exergy equation for a control mass is:

$$\phi = \int_1^2 \left(1 - \frac{T_0}{T_1} \right) \delta Q - (W - p_0 \Delta V) - I \qquad 4.2$$

where: T = temperature, K

Q = heat transferred, kJ

p = pressure, kPa

V = volume, m^3

I = irreversibilities, kJ

W = work transfer, kJ

$_1$ indicates current position

$_0$ indicates base condition

The main difference with the control mass assumption is within the enthalpy term where:

$$h_1 = u_1 + p_1 v_1$$

h = enthalpy, kJ/kg

u = internal energy, kJ/kg

p = pressure, kPa

v = specific volume, m^3/kg

In moving from point 1 to 2,

$$\Delta h = u_2 - u_1 + (p_2 v_2 - p_1 v_1)$$

For a constant pressure processes the bracketed term becomes $p \Delta v$. This term represents the work that can be done by the fluid in expanding. But in the pump application only a portion of this pressure may be used to do useful work. The rest is doing work (increasing in volume) on the atmosphere, which (for that particular process) is not useful work. This work can be considered as "stored" by the atmosphere and is able to be used at a later stage. The stored energy is recoverable when the atmospheric pressure does work on the pump to decrease the volume of the vapour chamber. This process is examined in more depth in Appendix 4.

The exergy values for the n-pentane at points 1 to 5 are given in Table 6. Given the exergy values at each point relative to the base condition, an analysis can be carried out to estimate the efficiency of each process.

Table 6. Exergy Values

Point	Exergy kJ/kg
1	0^- [1]
0	0
2	0^+ [1]
2'	17.8
3	44.7
4	28.8
5	8.9

[1] These values are very slightly positive and negative but are impossible to calculate given the accuracy of the enthalpy and entropy data for n-pentane.

It was very difficult to calculate the exergy value of the n-pentane from tabulated values of enthalpy and entropy (Canjar, 1967), due to the lack of accuracy to which these numbers are known. This problem occurred in the calculation of change of exergy in going from a saturated liquid to a saturated vapour. For water and some organic liquids, the change in exergy for this process given a control volume was zero, but for the pentane an erroneous positive value resulted. The reference from which the values were taken stated that entropy values had an accuracy of approximately plus or minus three percent. The problem with this is that to calculate the exergy, the change in entropy value is multiplied by the base temperature (in the order of 290 K). This has the effect of multiplying the magnitude of the 3% inaccuracy many times. The final process in working out the exergy is to subtract the enthalpy change from the entropy change multiplied by the base temperature. This leads to the classic case of subtraction of two equivalent sized numbers, leaving the errors in the calculations and original numbers drastically increased in size and in this case, dominant. In order to overcome this, the maximum amount of work that could be done by the pentane in going from one point to the next was computed from first principles and the resulting number was taken as the exergy of the pentane at that point.

4.2.1 Solar Radiation to Heat

The exergy value of radiation has been investigated in several theoretical studies, but few of them have reached agreement on the exact conversion efficiency of solar radiation. There are two main areas dealing with incoming radiation: the first is the computation of the maximum conversion efficiency of radiation assuming a Planckian radiation distribution. The second area deals with equating the Planckian distribution to the incoming radiation which has been absorbed in certain spectra by CO_2, O_3, H_2O, etc. Scattered, diffuse and reflected components also exist and their exergy can be calculated.

4.2.1.1 Maximum conversion efficiency of solar type radiation.

There are four main fields of thought on the conversion efficiency, each of which gives a different result (Jeter, 1981). The basis for the analysis is a system which comprises a volume of radiant energy, which after having completed a reversible process, is at the base or atmospheric state. The equation derived by Gudkov et al. (1986) and Gribik et al. (1984), differs only slightly from that derived by Landsberg (1979), Petela (1964) and Press (1976) who use one less simplifying assumption than the former. Giving

$$\max \eta = 1 - \frac{4}{3}\frac{T_o}{T_s} \qquad 4.3$$

and

$$\max \eta = 1 - \frac{4}{3}\frac{T_o}{T_s} + \frac{1}{3}\left(\frac{T_0}{T_s}\right)^4 \qquad 4.4$$

where:

A = cone half angle of incoming radiation

T_o = base atmospheric temperature, K

T_s = sun radiation temperature, K

these two equations differ numerically by approximately 0.0003%. The third equation

$$\max \eta = 1 - \frac{4}{3}\frac{T_o}{T_s}(1 - \cos A)^4 + \frac{1}{3}\left(\frac{T_0}{T_s}\right)^4 \qquad 4.5$$

was later amended by Parrot (1979) to

$$\max \eta = 1 - \frac{4}{3}\frac{T_o}{T_*} + \frac{1}{3}\left(\frac{T_0}{T_*}\right)^4 \qquad 4.6$$

where:

$$T_* = \frac{T_s}{(1 - \cos A)^4}$$

The final equation was suggested by Jeter (1981) as:

$$\eta = 1 - \frac{T_o}{T_s} \qquad 4.7$$

The form of equation to be used later shall be equation 4.3 . It has been suggested by Gibik (1984) that the equations 4.4 and 4.5 merely reduce the radiation to an intensity at T_o. In reality, the radiation is totally destroyed as is assumed in the derivation for equation 4.3 . Equation 4.6 takes into account the angle of incoming radiation, but this has been disputed by Wexler (1979). The final equation assumes that radiation is a flow process and that flow work of radiation should be taken into account. This is difficult to believe, because radiation does not interact with other radiation and therefore flow work is highly improbable.

4.2.1.2 Effective radiative temperature of solar radiation on earth

Having decided on the equation by which to analyse the radiation, there is a need to decide upon T_s (the upper temperature used in the equation). It must be taken into account that there is direct and diffuse radiation and that certain wavelengths have been absorbed by the atmosphere's gases.

The upper temperature T_s, which it is desired to find, is the temperature of a black body which has the same radiation energy output as the radiation energy received by the earth from the sun. Some of the studies for conversion efficiency which were reviewed above, went on to predict the efficiency for diffuse and actual direct solar radiation. Both Landsberg et al. (1979) and Press (1976) suggest, that for diffuse radiation, there is a 25% drop in conversion efficiency. Gudkov et al. (1986) assumes that the background radiation is at 300 K and therefore has a negligible effect on the conversion efficiency. The most realistic study was undertaken by Shafey et al. (1990), who analysed 62 wavelengths spaced at equal intervals, integrating over these wavelengths after diminution for the spectral absorbtion of gases. The paper then works backwards to find the equivalent black body temperature to be used in the Carnot efficiency equation for the remaining radiation. Their work assumes that the maximum possible work is by the Carnot cycle and associated efficiency, not equation 4.3 as was decided upon in the previous section. Despite this discrepancy, the work gives good information on the three components of radiation and only slightly erroneous effective temperatures. A final study by Haught (1984), calculates the radiation input to a collector also, but then goes on to consider radiation losses from the collector. From an energy balance the temperature of the surface is found and the Carnot cycle is used to calculate the maximum work that could be done. The radiation loss from the collector was assumed only to be re-radiation from the top surface.

4.3 EXERGY RESULTS AND PROCESS EFFICIENCY

From a knowledge of the processes undergone in the cycle and the intermediate states of the working fluid between the processes, the exergy and work output can be equated for each process. The calculations give the Second Law efficiency of each process and thus an overview of the processes which are most deserving of improvement.

4.3.1 Solar Collector

Owing to the standard flat plate collector's characteristics, it operates at a temperature to produce vapour at 68 °C and has a first law efficiency of approximately (Kishore, 1984):

$$\eta = 0.6625 - \frac{3.98 \; (T_p - T_a)}{I}$$

This equates to 46% for the pump's assumed operating conditions. Using a 2.9 m^2 collector and assuming solar radiation at a level of 1 kW/m^2, a total of 1.334 kW is transferred to the working fluid. From the previous section, the availability from radiation was estimated as equation 4.3 with the upper temperature being approximately 3,600 K. This gives a maximum conversion efficiency of 85.6% and an exergy of 2.482 kJ/s. The working fluid enters and exits the collector at the rate of 0.0015 kg/s, so solar radiation impinges with an exergy of 1650 kJ/kg of working fluid. The fluid enters with an exergy of 0 kJ/kg and leaves with 44.7 kJ/kg, giving a very low second law collector efficiency i.e. 44.7/1650 = 2.71%.

4.3.2 Expansion of Vapour Chamber

As discussed in Appendix 4, the ideal expansion process would take the vapour from 17.8 kJ/kg to 44.7 kJ/kg. The gain indicates 26.9 kJ/kg of work is available to be done in this process. This would be realisable if the water was being expelled with no losses, i.e. no acceleration of the fluid, no pipe or fitting losses, no condensation on the vapour chambers walls and no dead volume in the pumps vapour chamber (all of which can be considered irreversibilities). In reality with a working fluid flow rate of 0.0015 kg/s or 0.015 kg per cycle, the exergy in this process is 0.404 kJ/stroke. For a 3 litre stroke the ideal process would raise the water 13.4 m. In reality, this cycle raises the water up 6.5 m, indicating a 48% second law efficiency for the real pump in this process.

4.3.3 Unresisted Expansion

The unresisted expansion process produces no work, yet it allows the vapour to go from 44.7 kJ/kg to 28.8 kJ/kg. This process is the most wasteful within the pumping cycle. It is usual that this expansion process produces the power in the Rankine cycle. In this cycle it is wasted potential, i.e. 15.9 kJ/kg or 0.24 kJ/stroke is wasted.

4.3.4 Constant Volume Condensation

This process also produces no useful work and yet the exergy goes from 28.8 kJ/kg to 8.9 kJ/kg indicating as above a 19.9 kJ/kg loss of potential work.

4.3.5 Constant Pressure Condensation

This process is similar in theory to the expansion process. It starts with vapour at 8.9 kJ/kg. After the atmosphere has done work in compressing the vapour and energy has been extracted from it, it has zero exergy. The equivalent work that it could perform on the 3 litres of water is to raise it 4.45 m. In the actual pump it is only raised 1.87 m, giving this process a 42% second law efficiency. The losses in this process comes about by: accelerating the water out of the pump, pipe and fitting losses, condensation on the vapour chambers walls and by using more than 0.0015 kg/s due to dead volume in the pump which is condensed giving no extra work output (all of which can be considered to be irreversibilities).

4.3.6 Constant Pressure Increase in Temperature

There is no work extracted during this constant pressure process as it is the heating stage where the liquid condensate is taken up to boiling point in the solar collector. The initial exergy of the condensate entering the collector is zero and after heating, has an exergy value of 17.8 kJ/kg.

4.4 DISCUSSION AND CONCLUSIONS BASED ON EXERGY

From an analytical point of view, the exergy analysis should have been carried out at an earlier stage, when the initial concepts were being assembled into a first concept pump - collector - prime mover combination. At this later stage one can only note the effects that certain irreversibilities have on a process and their magnitude should they be able to be improved.

The work that is performed by the prime mover in the cycle described is not carried out in the process that a Rankine cycle would usually perform work. The cycle is rather unconventional and is significantly less efficient than the standard Rankine cycle. The work is done in the condensing and boiling stages. The Rankine cycle performs work where this cycle has an unresisted expansion. It is worthy of note, that the process in this cycle where work is done, is only possible because of the prime mover's interaction with the atmosphere. In the Rankine cycle this work in the boiling and condensing stages is unavailable.

The three processes which involve very low Second Law efficiencies can be altered to some extent but practical considerations rule out most of these possibilities. The two processes 3 to 4 and 4 to 5 (being the unresisted expansion and the constant volume condensation) could be replaced by an expander. This would extract some of the energy which is being wasted in the unresisted expansion. This work could power a feed pump for the condensed working fluid or even drive a second type of water pump. Practical considerations rule out this possibility as the unit would need a rotating shaft to utilise the available vapour, and as such, has been ruled out in Chapter 3 because of maintenance and manufacturing concerns.

It is also possible to increase the second law efficiency of the solar collector by concentrating the solar radiation and thus enabling a higher temperature and greater efficiency conversion into work. Such collectors would need to track the sun to operate for a whole day and generally would be too expensive and involve technology that is too complex for the desired application.

The two isothermal power producing processes in the pumping cycle have moderate efficiencies[1] of 48% and 42%. The irreversibilities both internal and external to the prime mover which cause the low efficiencies have been pointed out as flow losses, acceleration losses, unwanted condensation in the vapour chamber and dead volume in the vapour chamber losses. Both the flow and acceleration losses can be reduced by enlarging the inlet and outlet water pipe diameters. This is limited by practical considerations because the piping in use is already 40mm in diameter and has a low flow velocity through it. Attention to details within the vapour chamber could see some of the dead volume reduced and thus a reduction in vapour needed per pump stroke. Improved lagging would also reduce the vapour needed per stroke by reducing the condensation on the vapour chamber's walls. The combination of the above considerations could see an increase in water output per unit of vapour and therefore it would be possible to reduce the size of the solar collector, enabling a reduction in the total capital cost.

[1]This efficiency is defined as the work output of the process divided by the exergy change of the process.

5 COMPUTER OPTIMISATION AND SYSTEM MODELLING

Given the basic design and desired configuration of the pump, collector, prime mover and ancillary components, it is possible to evaluate the trade-offs between efficiency, size and their associated costs. It is possible to make the decision on the operating temperature of the system as in this chapter the temperature's relation to the pump's dimensions and operating conditions can be assessed. The effect of atmospheric conditions (i.e. location, cloudiness, radiation and ambient temperature) and pumping head on the operating temperature, diameter ratio, stroke and the final performance figures are also evaluated.

5.1 THEORY OF MULTI VARIABLE OPTIMISATION

There are numerous methods of optimisation for both single and multiple variables. The multi-variable optimisation methods are typically more complicated extensions of single variable routines.

Optimisation, in general, takes the model of a situation and tries to find maxima or minima given certain restraints and conditions. The particular variable or set of variables with a weighting placed on them, which it is desired to minimise, is the objective function. The objective function can be defined in terms of independent variables. These independent variables might be bounded, restricted or be part of an equation which is bounded or restricted. Of all the possible combinations of variables, there is a region within the search area which is valid i.e. all variables are within their bounds. It is probable that within this region there is a maximum, but there could also be local maxima of lesser value than the global maximum. In order to try to find the global maximum there are numerous methodologies, which depend on the type of objective function to be solved. Some methodologies converge on the solution faster than others and some are more prone to problems or obtaining non-global maximum points.

5.1.1 General Optimisation Procedures

The most basic optimisation procedure is to divide each variable into several equi-spaced points over its range and compute the outcome for each of the possible combinations of variables (cases). On noting the value of the variables at the most desired position, the optimum is found. This method is very inefficient and is only of use when the number of cases is small compared to the computational process and speed available. The more complex methods operate either on the objective function or on the output from the objective function. The latter approach enables optimisation of a system where the objective function may be derived from an iterative routine. When optimising linear or non-linear objective functions, the basic ideology is to reach the global maximum by noting the variable of greatest sensitivity within the equation (by differentiation) and then moving along this variable by an amount determined by a one dimensional search on this variable (eg. Golden Section search, gradient method). If the objective function can not be differentiated algebraically, a numerical method of differentiation can be used. It is the numerical method which forms the basis of the water pump's optimisation.

5.1.2 The Optimisation Routine Used

The objective function to be optimised in this program relies on an iterative routine. To enable solution, the optimisation procedure uses numerical differentiation to find the variable of greatest sensitivity. This variable is then increased or decreased to produce the desired response. The process then repeats and the next variable of greatest sensitivity is found. The numerical differentiation is usually performed by altering each variable by a very small amount and gauging the gradient from this change. However, for the iterative routine used in this program, the variable altering needed to be large to get an accurate change in the iterative procedure's output.

5.2 MODELLING OF THE WHOLE SYSTEM

In the combined pump-prime mover-collector unit (discussed in the previous chapters) it is possible to model the set-up completely with a computer program.

The initial step was to model the influences on the unit for any particular location, from this point any modelled pumping unit could be inserted to find the expected performance for any location, at any time on any typical day of the year.

The program was based on Kysar's (1988) program which was set up to track sunlit areas through rooms. Kysar's program ended up being totally re-written to avoid the accurate yet cumbersome sorting of inputs. Only five small subroutines from the original program remain in the final version. The errors with Kysar's program, of the sun tracking in the wrong direction for the southern hemisphere and an error in directional coordinates of the incoming radiation were rectified. The complete computer program is printed in Appendix 5.

5.2.1 Solar Angle

The angle of the incoming solar radiation is calculated in terms of u,v and w, which are the directional cosines of the incoming radiation and are related to the azimuth and altitude angles. The azimuth and altitude angles are dependent on one's latitude on the earth, time of day and declination (related to the season). The subroutine SUNAG used to perform these calculations is adapted from Kysar (1988) (where a more complete discussion can be found).

5.2.2 Radiation Intensity

The radiation intensity taking into account the earth's elliptical orbit is based on the radiation look-up table and equations presented in ASHRAE (1989).

$$I = \frac{A}{e^{\left(\frac{B}{\sin Alt}\right)}} \qquad 5.1$$

where:

A = apparent solar radiation at air mass = 0, W/m^2

B = atmospheric extinction coefficient

I = insolation, W/m^2

Alt = solar altitude

This equation is represented in the computer program as an interpolating section within the main program, which plucks the A and B values from a look-up subroutine COEFF and adjusts for the day of the month. The final coefficients are passed from the subroutine into the main program where they give a relevant radiation intensity for a clear day. The radiation intensity value found is an average value for a cloudless day. For a very clear day the radiation intensity can be up to 15% greater than the equation predicts. For an air mass of over one or semi cloudy conditions, the radiation will be less than that predicted ('air mass' is the mass of air, dust, etc. between the sun and earth).

5.2.3 Glass Absorption, Reflectance and Transmittance

The values for absorption, reflectance and transmittance of light interacting with a glass cover were calculated, given the relative angle at which the radiation strikes the glass (Shurcliff, 1974). The transmittance τ, is given by:

$$\tau = \frac{1}{2}\left(\frac{1 - R_p}{1 + (2N - 1) R_p} + \frac{1 - R_s}{1 + (2N - 1) R_s}\right) \qquad 5.2$$

where:

$$R_p = \frac{1}{2}\left(\frac{\tan^2 (i - i')}{\tan^2 (i + i')}\right)$$

$$R_s = \frac{1}{2}\left(\frac{\sin^2 (i - i')}{\sin^2 (i + i')}\right)$$

i = angle of incidence

i' = angle of refraction

N = number of glass sheets

$_p$ indicates in plane of incidence

$_s$ indicates perpendicular to plane of incidence

For the indirect or diffuse components of radiation, the transmittance has been related to a hypothetical angle of incidence (Brandemuehl et al., 1980). The angle gives an effective beam radiation incidence angle to be placed in the above equation ie. for sky diffuse radiation the incidence angle is:

$$i^e = 59.68 - 0.1388 \times B + 0.002693 \times B^2 \qquad 5.3$$

where B = collector slope in degrees

for reflected radiation from the ground

$$i^e = 90 - 0.5788 \times B + 0.002693 \times B^2 \qquad 5.4$$

The absorption of the glass is allowed for by multiplying the standard absorption by a factor in order to account for the path length of the light through the glass. The remainder of the radiation (once reflection and absorption have been accounted for) is transmitted to the collector's surface.

5.2.4 Cloudiness

Cloudiness is a measure of the total amount of sky that is covered by cloud (e.g. a cloudiness of 1 indicates overcast). There are three main ways to measure cloudiness. The most subjective, yet most common is the human estimation from a view of the sky. In general this method is moderately accurate but overestimates are made in certain conditions, due to the viewer seeing the side of clouds at low angles of inclination. The second and third methods eliminate the inclination error. The second is conducted by finding the cloud area ratio from satellite photographs. The third method uses a meter on the ground, which times the periods of cloudiness and then averages them over a given time period to give the cloudiness ratio. For this program it is assumed that all cloudiness values will be from ground estimates and thus a series of correction factors from Malberg (1973) for the human error can be applied. The factor is used to overcome the inaccuracy brought about by the person seeing the side of the clouds (Malberg, 1973). This correction equation has been calculated for various latitude observers and is a function of observed cloudiness and latitude. The effect of the corrections come into play gradually during the day until midday and then tapers off as dusk approaches. The variation in the correction is because the ground observation of the side of clouds is correct when this is the path of the sun's rays ie. in the morning and afternoon.

For observer correction CS = 0.965 cloudiness - 0.12 5.5

For latitude correction CC = CS + 0.24 -0.0018 x latitude 5.6

where CS is observer corrected cloudiness and CC is the observer and latitude corrected cloudiness.

5.2.5 Ambient Temperatures

The ambient temperature during a day's progress has been mapped by Trewartha (1980) for certain places at different latitudes. It was noted that from these isotherm maps, on any day, irrespective of the time of year, the range of temperatures remains constant. The temperature range also remains constant for the seasonal variation at different times of the day. From this starting point, given an average daily temperature range and an average monthly temperature range, along with the average maximum temperature for the hottest season, a locations average temperature during the day can be mapped. This is demonstrated in figure 25 where AA measures the daily temperature variation and BB measures the seasonal temperature variation. Note, this method is applicable to most locations but is very approximate and does not account for seasonal phenomenon such as monsoons, local anomalies, etc..

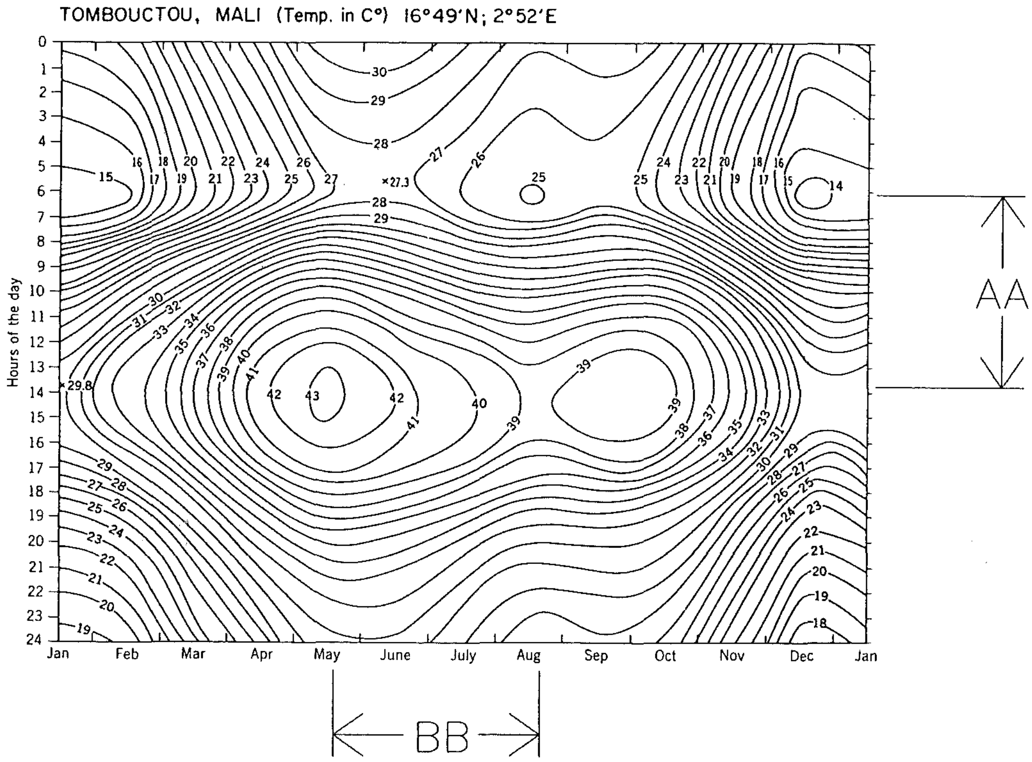

Figure 25. Daily Temperature Variations (from Trewartha, 1980)

5.2.6 Down Time

Down time is a rough measure of the period of time which the pump is not working due to clouds intercepting the direct radiation before it reaches the solar collector. The main phenomena which need to be taken into account is the increase in diffuse radiation which occurs as a result of the direct radiation now impacting upon the cloud and thus giving off more diffuse radiation. There have been several studies conducted on the diffuse radiation intensities relation to cloudiness and cloud type. Lestrade et al. (1990) gives a relationship between cloud cover and the average magnitude of direct and diffuse radiation. Robinson (1977) draws a correlation between the angle of scattering and the relative intensity of the scattered radiation during the period of cloud cover. Kimura (1969) also gives results for the diffuse - cloudiness interaction. The computer program corrects the viewer's cloudiness observation for both side viewing and latitude for each quarter hour period. A loop is then encountered to find the length of cloud cover during each quarter hour period. The period of cloudiness is then passed down to the WOKOUT subroutine. The temperature to which the collector will fall is calculated (given the calculated diffuse radiation) as was suggested by Duffie et al. (1980). The period of time needed for the collector to reheat, to working temperature is then calculated. The final down time as a percentage of the ¼ hour period, is subtracted from the work output of the pump which would have operated continuously over that quarter hour period.

The equation relating the change in temperature to solar radiation and time (Duffie 1980) is:

$$\frac{S - U_L (T_p - T_a)}{S - U_L (T_{pinitial} - T_a)} = e^{-\left(\frac{A_c U_L \tau}{(mC)_e}\right)} \qquad 5.7$$

where: S = insolation, W/m^2

U_L = top loss coefficient, W/m^{2}K

A_c = collector area, m^2

$(mC)_e$ = effective heat capacity, J/K

T = temperature, K

τ = time, s

$_c$ indicates cover

$_p$ indicates plate

5.2.7 Modelling of the Water Pump - Prime Mover

The modelling of a prime mover (which operates under the cycle described in detail in Chapter 4) would not be precise if the pump's ideal operating cycle is used (in a modified form) to predict the pumps performance. This method of modelling was not used because of the difficulty of accurately modifying the cycle to include inefficiencies in a form which could lead to an accurate solution over a day's pumping.

The prime mover operates basically as a displacing cylinder. The motion of the cylinder is produced by the pressure of the working fluid. It is possible to view the working fluid as a provider of pressure, whose density and energy content per unit volume is known. By working with pressures and volumes displaced (including allowances for dead volume, diaphragm deflections, limiting pressures and condensation) the system can be modelled as a combined heat-mass transfer system.

5.2.7.1 Model of water intake

The simulation operates as follows: with the diaphragm at the bottom of its travel, the valve opens the vapour chamber to the solar collector. The collector's temperature and thermal mass are known and are added to by the thermal mass, volume and energy of the vapour in the vapour chamber. The pressure in the solar collector is slightly reduced by the addition of the dead volume of the vapour in the vapour chamber. The pressure of the vapour in the collector is then calculated from a knowledge of the working fluid's boiling point at that particular temperature. The collector's pressure is then used to calculate the deflection of the diaphragm caused by the pressure change from condenser pressure before the valve opened to the collector's pressure. The deflection of the diaphragm displaces a certain volume of water, which is taken up by the initial motion of the diaphragm (no water is pumped). The pressure and temperature are re-calculated since some vapour has been boiled off by the collector to increase the volume of the vapour chamber. Now an iterative routine is entered into in order to simulate the travel of the diaphragm during water intake. The initial travel because of diaphragm deflection is known, as is the maximum stroke (from the calculation which limits diaphragm length due to pressure across it). The iterative routine takes the diaphragm from its initial travel to the maximum travel, as follows: from a pressure balance of the pump-prime mover there is a resultant force due to the high pressure of the collector/vapour chamber. This imbalance is dissipated over the area of the water chamber and accelerates the water from the bottom of the well through the piping, into the water chamber. The water velocity is adjusted for pipe, entry, fitting and valve losses. In the suction intake there is an upper limit placed on the accelerating pressure. This upper limit is the pressure at which the water cavitates (Brett, 1985) and is approximately 5 kPa absolute (dependent on the water's temperature). The increase in vapour chamber volume over a 1/20th second interval is related to the flow of water into the water chamber. This increase in volume, along with the condensation on walls and solar energy added, is calculated to give a new overall system temperature and pressure. This pressure is fed back to the start of the next iteration and the iterations continue until the maximum travel is attained.

5.2.7.2 Model of water delivery

On reaching the maximum travel the valve shuts off the collector and opens the condenser to the vapour chamber. A similar approach to that adopted above is used for the condensation stage, i.e. the temperature and thermal mass of the condenser is computed. The pressure in the vapour chamber is calculated from the combined energies of its vapour and the condenser's vapour's energy. A pressure balance on the pump can then be found. The excess pressure acting on the water chamber accelerates the water out of the pump through the same iterative routine as above. This routine stops when the diaphragm reaches the bottom of its travel. The time for this condensing portion of the cycle is then used to calculate the extra energy gained by the collector from the sun and thus its new temperature and pressure.

The output from the model is: water intake time, water delivery time, volume of water pumped and overall energy gained or lost during one cycle. The main features of this model are: the system is modelled by heat and mass transfers and volume changes, the model includes allowances for vapour condensation on the walls of the vapour chamber, dead volume losses in the vapour chamber, limiting pressures in the water chamber due to cavitation and diaphragm deflections under pressure. There are four losses which have not been accounted for in the simulation programme. The first is the force needed to deflect the rubber diaphragms and includes to some extent, losses due to hysteresis in the rubber. The second loss is the pressure loss of the vapour in travelling from the collector to the vapour chamber and from the vapour chamber to the condenser. It is possible that the tortuous route of the vapour would cause a pressure loss, but it is expected to be small due to the low density of the gas. The low density enables the gas to be easily accelerated. The large pressure differences driving the gas's motion ensures quick response in the transferral of pressure and in the moving of the gas from one point to another. The third loss is the fin type heat exchange which will occur in the pumps bottom and top plate to some extent. The final effect not accounted for is the small amount of vapour which goes directly from the collector to condenser while the valve is changing position and both ports are open for a short time.

5.2.8 Top Loss Coefficient

Numerous models have been proposed for the top loss coefficient of a flat plate solar collector ((Malhorta et al., 1981)(Agarwal et al., 1981)(Cooper et al., 1981)(Phillips, 1982). A comparison of the results with tests of an actual solar collector by Garg (1984) leads to the following equations:

$$U_L = \left(\frac{N}{\frac{c}{T_p} \left(\frac{T_p - T_a}{N + f} \right)^e} + \frac{1}{h_w} \right)^{-1} + \frac{\sigma \, (T_p^2 + T_a^2) \, (T_p \; T_a)}{\frac{1}{d} + \frac{2N + f - 1}{E_g} + g - N} \qquad 5.8$$

where:

$$c = 204.429 \, (\cos\beta)^{.252} / L^{.24}$$

$$d = E_p + 0.0425 \, N \, (1 - E_p)$$

$$e = 0.252$$

$$f = \left(\frac{9}{h_w} - \frac{30}{h_w^2}\right)\left(\frac{T_a}{316.9}\right)(1 + 0.091\,N)$$

$$g = 0.0$$

U_L = top loss coefficient, W/m^2K
N = number of glass panels
T = temperature, K
E = emittance
L = spacing between plates, m
β = tilt of collector measured from horizon, °
σ = Stefan Boltzmann constant, W/m^2K^4
h_w = glass to air convection coefficient, W/m^2K
$_p$ indicates plate
$_a$ indicates atmosphere
$_g$ indicates glass
$_w$ indicates glass cover to environment

The c,d,e,f and g coefficients are the portions of equation 5.8 which were varied in other models, those quoted here gave the best results when compared to an actual solar collector.

5.3 PROGRAM SET-UP

The external inputs into the pump model and the basic model of the pump need the program to be set up properly. The set-up is needed to ensure steady state operation of the model under the given conditions. The possibility of variable changes is also included in the program so their effect on the final performance of the pump can be gauged. Therefore it is possible to perform an optimisation of the unit's dimensions for any conditions.

In order to enable optimisation the program is designed to obtain steady state operation of the pump when given a particular input. The steady state output is then adjusted for cloudiness considerations. The results for a whole day's pumping with different variables changed are compared and the optimum pump based on total water output is chosen. This procedure is run through for different solar input levels. The best pump is then able to be chosen by selecting the pump parameters which produce the greatest water output for one days operation.

5.3.1 Maintenance of Steady State

For the pump to be simulated at steady state conditions the subroutine for simulation of the pump-prime mover-collector must return an equal amount of energy lost as is gained for any particular situation, i.e. an energy balance must be achieved within the collector/pump unit. In order to ensure the energy balance in each simulation run, the energy lost by the solar collector was calculated, i.e.:

$$M_v(C_{pg}\ T_g + h_{fg}) + U_p - M_L(C_{pf}\ T_f) = (A - \frac{B\ (T_g - T_a)}{I})\tau_c I \qquad 5.9$$

where:

M_v = mass of vapour, g
C_p = specific heat of the fluid, J/gK
T = temperature, K
h_{fg} = latent heat of vaporisation, J/g
M_L = mass of liquid, g
U_P = miscellaneous energy losses, J
A,B are collector constants
I = insolation, W/m^2
τ = cycle time, s
$_f$ indicates liquid
$_g$ indicates gas

The length of time for each cycle is noted and multiplied by the collector's efficiency and solar input. For an energy balance, the energy gain and energy loss values must be equal. A minimisation of the difference of these two values was undertaken by a Golden Section search working on collector temperature.

The Golden Section search thus finds the temperature and pressure at which the collector operates (if energy is gained over a cycle the temperature is increased, if energy is being lost then the temperature is decreased).

There are, however, several failure scenarios when the program is being used to optimise pump parameters. The failure scenarios have been identified and when one occurs, rectifying action is taken. The possible failure scenarios are:

A) for low temperatures/pressures the collector's vapour pressure may not be high enough to lift the diaphragm,

B) for large diameter ratios, the condenser may not produce a low enough vapour pressure to suck the diaphragm down,

C) not enough insolation reaches the solar collector to replenish the energy being extracted by the prime mover.

When the final steady state condition has been found for the system, the water output is altered by a certain amount to take into account the possibility of clouds. The cloudiness calculations are performed prior to the finding of steady state conditions of the pump so the actual time of cloud cover for every quarter hour period is known. This period of time for which the solar collector's direct radiation is diffused by cloud and no pumping occurs, is used to calculate the temperature to which the collector will drop (due solely to the collector's heat losses to the environment). The time needed to reheat the collector to within 10% of operating temperature by the normal amount of direct and diffuse radiation is calculated. This time is added to the cloudiness time to give a total down time or period of non-operation. The water which would have been pumped over this period of time is subtracted from that pumped over the quarter hour period. Therefore the final figure on which the pump's usefulness is based is the total water pumped (for a specified suction and discharge head) in a quarter hour period of direct solar radiation minus that which was not pumped due to cloudiness and recovery time.

5.3.2 Radiation Level Choosing

For simulation purposes, there is a need to vary the solar radiation input to try to influence some of the design parameters. There were two possibilities for an optimum pump over one days operation, depending on the solar radiation intensity at design conditions. Given the intensity of solar radiation for a cloudless day (Fig. 26) the two possible operating extremes are:

A) operation at low insolation levels for a long period of time, or
B) operation at high insolation levels for a short period of time.

System A would operate with say 400 W/m^2 insolation, which indicates that it would operate for up to 10 hours per day in ideal conditions. The low insolation level would encourage the solar collector to operate at a low temperature and thus have a high efficiency. Conversely the heat engine operating from the solar collector's output would have a low temperature heat source, causing it to be less efficient than an engine working from a higher temperature heat source. System B operates with say 850 W/m^2 insolation, indicating a running time of up to 6 hours for a cloudless day. The high insolation level will give the opposite situation to system A, ie. the solar collector will operate at a high temperature and low efficiency but the heat engine operating off the solar collector will have a higher efficiency due to the high temperature of its heat source.

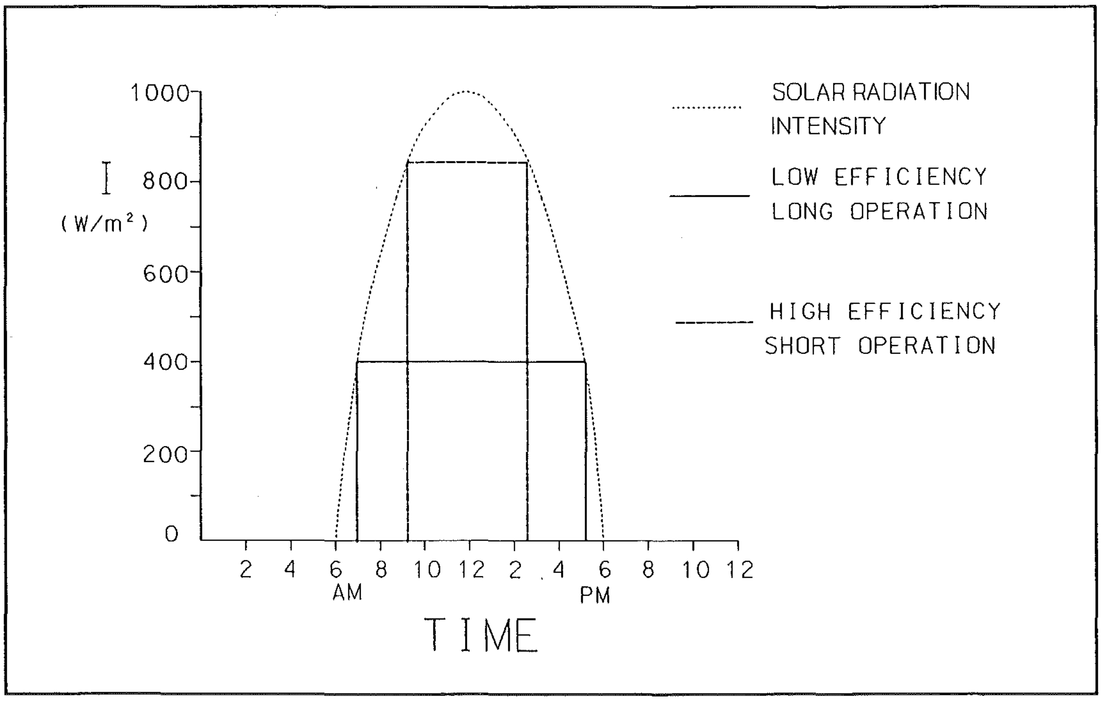

Figure 26. Possible Periods of Pump Operation

The choice of insolation forms the very outermost loop of the pump optimisation process. The pump is optimised for each insolation level. The work output and pump dimensions are given for each optimised solution and left in a file for the operator to identify the best solution based on total water output.

The insolation values tried are the maximum and minimum solar radiation minus and plus 150 W/m^2 respectively and then four points equally spaced in the range between the upper and lower trial insolation values.

5.3.3 Dimension Choosing

The dimension choosing routine will try to alter each of the dimension variables, to produce an optimum pump given initial dimensions which are viable. The initial dimensions can be varied between the insolation runs described above. The ability to vary the initial dimensions is necessary because the start point for the high insolation runs may not be viable for the lower insolation level pumps, as is suggested via the failure scenarios in Chapter 5.3.1.

The method of altering the variables to find the optimum was similar to the best gradient method, i.e. the subroutine alters each variable by +/- 20% of its initial value and performs one day's pumping in half hour steps. If any of the runs with an altered variable puts out a better work output than its predecessors, it will become the basis of the next set of variables for variation, again with +/- 20%. Should there be no improvement with the 20% variation, it is assumed that a maxima is within the 20% radius and the variation is reduced in 5% steps until another maxima is found. After reaching 5% variation the final step is 2½% variation. The final program operates as shown in Fig. 27.

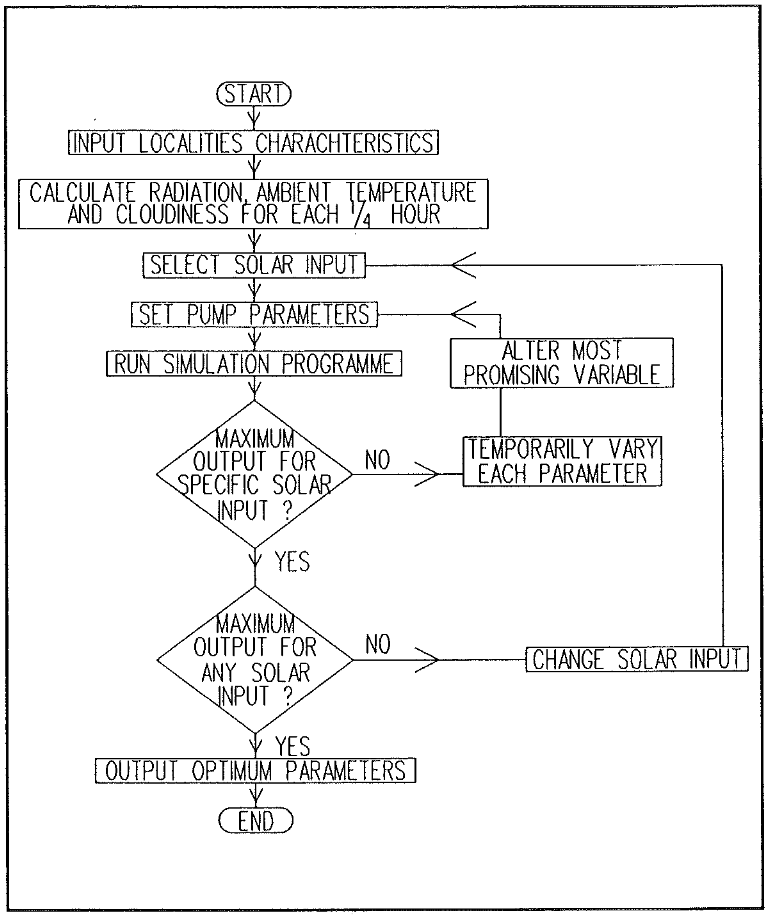

Figure 27. Optimisation Flow Diagram

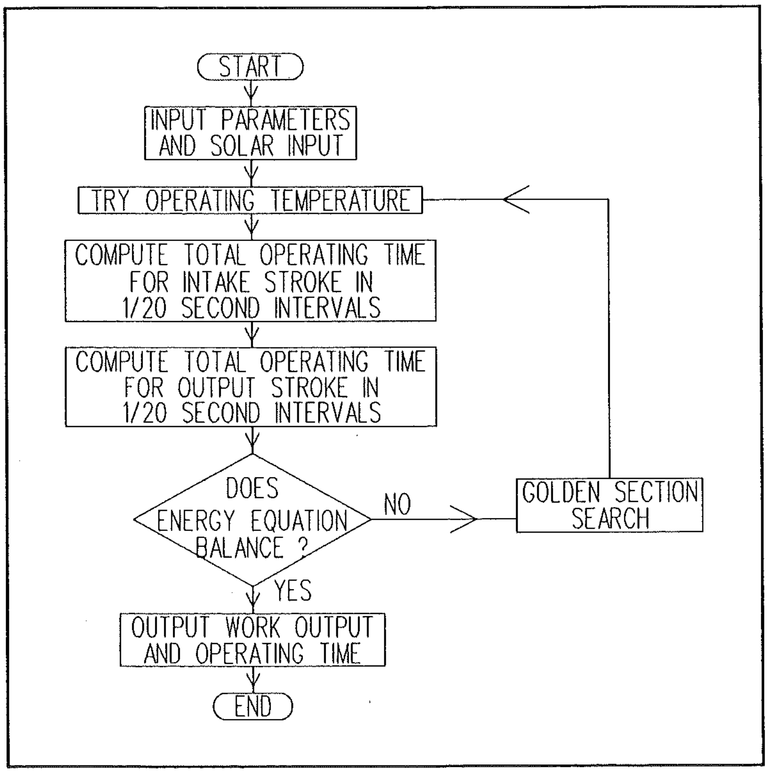

Figure 27.cont. Simulation Flow Diagram

5.4 RESULTS AND PREDICTED PERFORMANCE

The final computer program under trial, produced several results which were extreme in size. These results served as indicators of the functional nature of its optimised choices. The results were compared with theory from Chapter 4.

The variables of pump diameter, condenser volume, condenser area, diameter ratio and pump stroke were all free to be altered by the program initially. After optimisation it was found that the condenser volume and the condenser area tended to infinity. This tendency is predictable because there are no functional disadvantages due to large condenser areas and volumes. The large values were taken as a healthy sign that the variable altering and choosing routine was working properly. The condenser area and volume were then set at constant values which were within practical, financial limits.

For large insolations, the pump's diameter tended to large values, for lower insolations the diameters chosen were acceptable, so an upper limit had to be placed on the pump diameter to keep the unit's size practical.

A natural upper limit was placed on the stroke of the pump by diaphragm considerations. For a certain pressure acting across the diaphragm, a maximum length to ensure the diaphragm was not over stressed was ascertained. From knowledge of this length and geometric constraints the maximum stroke was computed.

Given a pump with a design insolation of 750 W/m^2; from pumping results which were put into files from the different solar insolation values, it was evident that the optimum pumps designed for insolation values greater than 750 W/m^2 were of similar diameter and stroke (given that some dimensions were constrained). For optimum pumps designed for insolations below 750 W/m^2, both the pump's total daily water output decreased and the pump's diameter and stroke decreased. At the design insolation the optimum pump-collector-prime mover had been found. However as the values of insolation dropped the dimensions needed to change to allow steady state operation. For higher insolations, the design insolation (optimum) was an option, and was chosen. This also was taken as a healthy sign that the radiation level was having the desired effect on the choice of pump dimensions.

The final proof of good operation of the program was obtained by the placement of the operating temperature of the optimised water pump back onto the graph of combined Carnot and collector efficiencies. This combined efficiency graph is shown in Fig. 28.

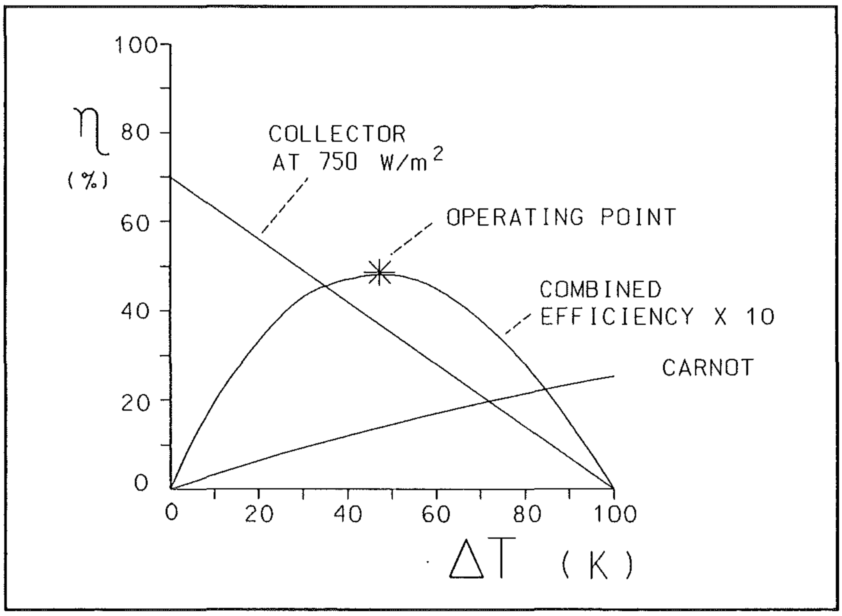

Figure 28. Operating Point

The actual choice of design insolation gives a pump whose operating point is at the start of the plateau of highest combined efficiency of the collector - Carnot pairing. Note, the plateau still maintains its form if the actual prime mover efficiency is say ¼ the Carnot efficiency, it will merely scale the height of the resultant line down by a factor of 4. By designing the system to operate at a temperature which is at the start of the maximum efficiency plateau, the pump's output will increase with an increase in solar radiation. This is due to the overall efficiency of the system remaining the same when the operating temperature increases and the energy input into the system increases. If the pump system was designed to operate at a temperature corresponding to the end of the maximum efficiency plateau, an increase in operating temperature would reduce the overall system efficiency and water output despite the increase of energy entering the system. Due to the pump spending most of its time at an insolation greater than the design insolation, the choice of an operating temperature which corresponds to the start of the maximum efficiency plateau would give a greater daily water output than the choice of operating points which relate to a higher design operating temperature.

5.4.1 Effects of Individual Variables

If changes were made to each of the design conditions and the corresponding changes in the pump's performance were found, it would be possible to find the sensitivity of the optimum pump set-up to certain variables. The variation of:

* pumping head,
* suction head,
* cloudiness, and
* altitude or seasonal variations

were investigated. The effect that varying the pump parameters has on performance was also investigated.

5.4.1.1 Pumping head and suction head

The optimisation program was run through several different head configurations. The optimal pump parameters were found and the output for an ideal day was plotted against pumping head of water (Fig. 29). The effect of varying the water delivery height above the pump unit was minimal. Because this portion of the cycle is dependent upon the condenser temperature (related to the temperature of the water being pumped) and has no direct bearing on the collector's temperature or pressure, small changes in delivery head produced minimal changes to the pump's output. The change in well depth altered the optimum diameter ratio and diameters to a moderate extent. In general the operating pressure and temperatures increased with decreasing well depth. The optimum results are represented by the peak in the m^4 versus head graph (Fig. 29). The m^4 or hydraulic energy equivalent performance (where $m^4 = m^3$ (volume of water pumped) $\times$ m (pumped head)) is a measure of the work output of a pump unit. To perform the same amount of work over different head conditions the ideal device would have a constant hydraulic energy equivalent line.

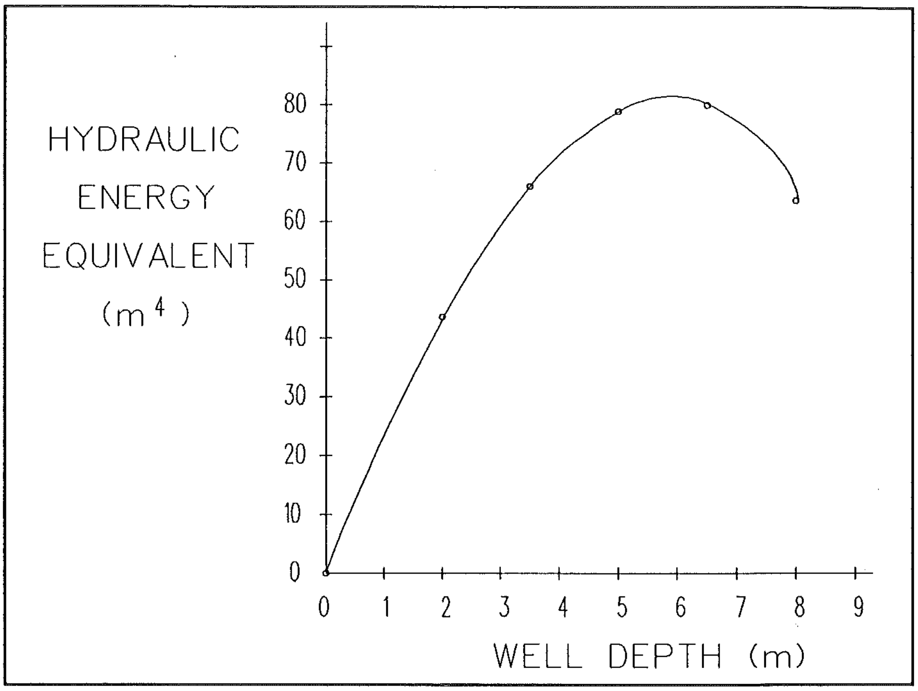

Figure 29. Pump Performance

The drop in energy equivalent for the 9.5 m pumping head (8m suction), is due to cavitation arising in the intake during the suction stroke. This reduction in performance is not caused solely by the water being displaced by vapour, but also by the vapour pressure during cavitation limiting the accelerating pressure for the water. Thus the water's velocity into the water chamber was also limited. It is possible for the vapour cavity to collapse at the end of the pump intake stroke, lifting the foot valve and replacing with water what would otherwise be volume occupied by vapour. This replacement of the cavitation volume is possible because of the slow motion of the diaphragm and slight pauses at the top and bottom of each stroke. The drop off in hydraulic energy equivalent for the shallower well depths, is because higher water flow rates cause the parasitic losses to be much greater, despite fluid friction factors remaining essentially constant (ie. same foot valve and pipe diameter). The flow losses can be overcome by using larger diameter pipes, fittings and valves where appropriate.

5.4.1.2 Cloudiness effects

The inclusion of a period of down-time to allow for the effect of cloudy periods was only of major consequence in the results for the simulations which operated at low levels of direct solar radiation. In this scenario, systems which have higher operating temperatures and greater thermal masses would lose more energy than systems with lower thermal mass and a lower operating temperature. In the final analysis this mode of operation was not of major consequence to the results because of the following: on the cessation of direct radiation, the system performs several pumps until the pressure in the solar collector is not sufficient to displace the lower diaphragm of the pump. At this point there is no vapour (energy) being extracted by the pump unit and thus the solar collector will go into temporary stagnation conditions. For cloudy intervals, the diffuse radiation can rise from its normal level with no cloud cover of approximately 1/10 of the direct component, to between 1/7 and 2/5 of the global radiation depending on cloud cover (Lestrade, 1990). For a collector operating at stagnation conditions under such diffuse radiation, the stagnation temperature is close to, if not higher than the unit's operating temperature in direct sunlight. Therefore the solar collector will stagnate at just below operating temperature. On the recommencement of direct solar radiation the system will start up with only a small time delay.

5.4.1.3 Monthly and latitudinal variation

The affects of latitude and declination on the pump - solar collector unit effects the choice of pump set-up and size to a large extent. For a location in Christchurch, New Zealand at latitude 43° south, the solar insolation would very infrequently reach over 700W/m^2 at the peak of winter. This indicates that a pump with a design insolation of 750W/m^2 will not work at all in the depths of winter, but may supply sufficient water in summer. The computer program only gives the optimum pump for a day's operation on the specified day in the specified month. To find the ideal pump for any latitude, the user must supply the desired water output for the worst month (i.e. highest water demand or lowest insolation or both). Generally, the program responds to an increase in the latitude or declination by decreasing the pump diameter and stroke for a constant collector area and maintains the other variables close to constant.

5.4.1.4 Effect of variation of pump parameters

The varying of the pump's diameter, stroke or diameter ratio individually will have an effect on the output of the pump unit. A variation of plus and minus 10% in steps of 5% was undertaken for these three variables. The results are presented in Table 7. From the results, it can be seen that the rate of output of the pump is most sensitive to changes in the diameter ratio. The other two variables give mixed results, some of which will be ruled out in the optimisation program. For example to get the results in Table 7 the maximum diaphragm stress and the maximum pump diameter limits were overridden. The output for these different pump set-ups were calculated only for the design conditions. Over a full days pumping, the ranking of the units may change from that in Table 7 due to operation at above and below design conditions. The pass-on effects from each variable can be estimated but the magnitude of the combined effects are best left to the computer program.

The increase of stroke was limited by the maximum pressure that could be withstood across the exposed area of cupped rubber diaphragm. An increase in stroke also elevates the heat losses in the pump unit across the uninsulated diaphragm whose area increases with stroke. The stroke increase also detrimentally increases the dead volume in the pump unit's vapour chamber (due to the deflection of the diaphragm). An increase in the diameter ratio between water and vapour chambers gives a relatively lower area of uninsulated diaphragm and beneficially increases pumping cycle time, but detrimentally increases the vapour chamber's dead volume. An increase in diameter ratio induces higher working fluid temperatures. These higher temperatures add to thermal losses through the uninsulated diaphragm and also lower the collector's efficiency, but give higher possible pumping cycle efficiencies. An increase in diameter ratio also makes the pump unit cycle slower, because more water is being induced with less pressure available to accelerate the water into and out of the pump's water chamber.

Table 7. Influence of Pump Parameters

Parameter	Parameter variation	Output m^3/stroke	Pumping time sec/stroke	Operating Temp. °C	Output l/s
Lower Diaphragm Diameter	- 10%	.00261	5.2	67.6	0.504
	- 5%	.00287	5.5	67.6	0.522
	base	.00318	6.1	65.7	0.524
	+ 5%	.00352	6.4	65.4	0.531
	+ 10%	.00387	7.1	64.7	0.546
Stroke	- 10%	.00289	5.5	67.6	0.526
	- 5%	.00304	5.7	67.5	0.532
	base	.00318	6.1	65.7	0.524
	+ 5%	.00333	6.3	65.3	0.528
	+ 10%	.00347	6.6	64.5	0.522
Diameter Ratio	- 10%	.00120	4.5	59.0	0.268
	- 5%	.00145	5.2	63.9	0.28
	base	.00318	6.1	65.7	0.524
	+ 5%	.00371	9.4	64.7	0.396
	+ 10%	.00426	14.7	67.5	0.29

Pump's base dimensions: Lower diaphragm diameter 0.27m, Stroke 0.02m, Diameter ratio 1.71

6 PERFORMANCE OF THE PUMP UNIT

Having built a prototype pump unit to the dimensions suggested by the dimensional optimisation in the previous chapter, it was desirable to test the unit and compare its performance with the computer simulation.

6.1 EMBODIMENT OF DESIGN

The optimisation program gave the optimum dimensions of a water pump, which pumps for a full day, in summer, in Christchurch, New Zealand. The final dimensions were: lower diaphragm diameter = 0.25m, diameter ratio = 1.67, diaphragm length = 20mm, stroke = 30mm.
Using these dimensional requirements the prototype design for the pump unit was created. The pump is designed to operate with: heat input = 660 W (insolation of 750 W/m^2 over 2.9m^2), water suction head = 6.5m, water discharge head = 1.5m, cycle time = 6.5 seconds, water mass flow = 2.5 kg per stroke.
The design took the form depicted in Fig 30.

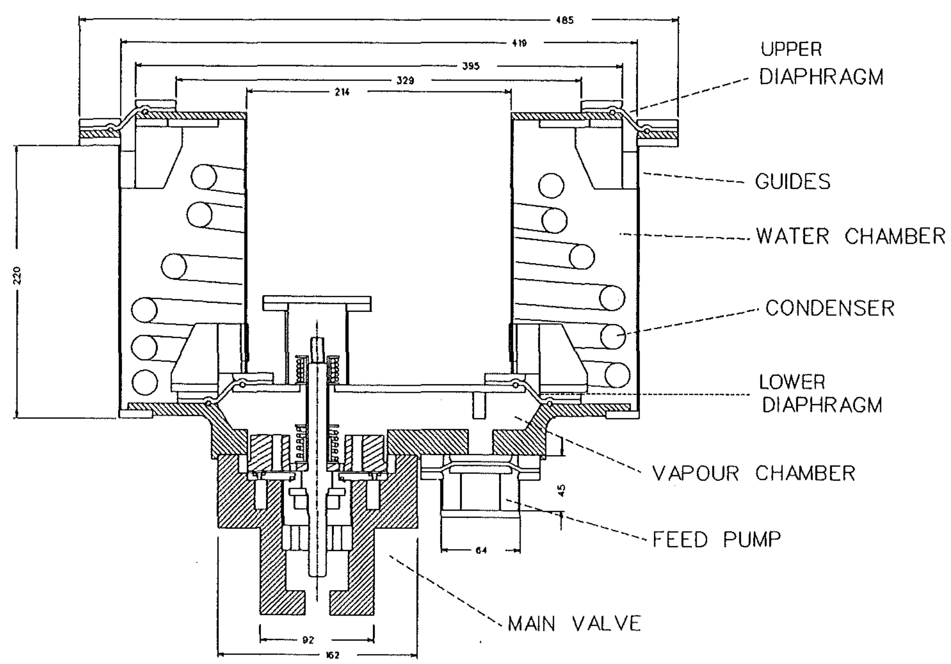

Figure 30. Prototype Design (all dimensions in mm)

6.1.1 The Condenser

The condenser was made from 5 metres of 1" diameter copper tube, spiralled into two small and three large coils (as can be seen in Fig. 31). The coil is immersed in water at a temperature of 17 °C. Pentane vapour condenses inside the coil at a temperature of 68 °C.

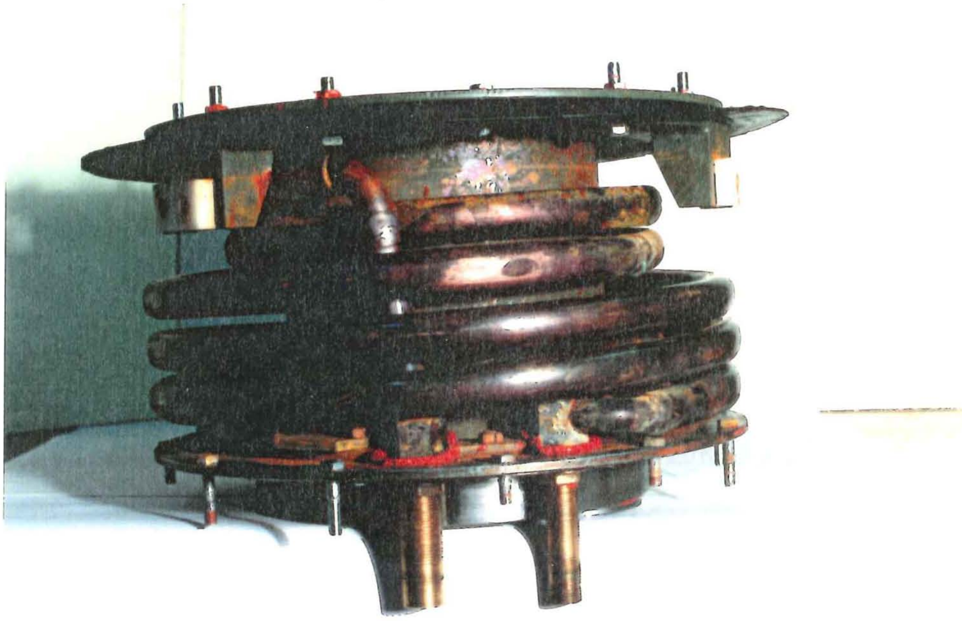

Figure 31. Pump without Water Jacket, Showing Condenser Coil and Guides.

Figure 32. Nitrile Diaphragms after Curing into Cupped Shape

For a horizontal tube with vapour condensing inside it the Chato equation is (Janna, 1986):

$$\overline{h_D} = 0.555 \left[\frac{g \, \rho_f \left(1 - \frac{\rho_g}{\rho_f} \right) k_f^3 \, h'_{fg}}{\nu_f \left(T_g - T_w \right) I D_t} \right]^4 \qquad 6.1$$

where:

$$h'_{fg} = h_{fg} + \frac{3}{8} C_{pf} \left(T_g - T_w \right)$$

h_{fg} = latent heat of vaporisation, kJ/kg
T = temperature, K
g = gravitational constant, m/s^2
k = thermal conductivity, W/mK
C_{pf} = specific heat, kJ/kgK
A = area over which heat transfer occurs, m^2
ID_t = tube internal diameter, m
ρ = density, kg/m^3
v = specific volume, m^3/kg
h_D = heat transfer coefficient, W/m^2K
$_g$ indicates gas
$_f$ indicates liquid
$_w$ indicates wall

This gives h_D the value of 891 W/m^2K. Putting this in the heat transfer equation

$$q = A \, \overline{h_D} \left(T_g - T_w \right) \qquad 6.2$$

with the heat transfer area equal to $0.032m^2$, the amount of energy able to be handled by the condenser is 1.45 kW. The condenser is only used for approximately 40% of the time, during this interval it must condense the pentane boiled off by the collector in ten seconds. If the collector works at approximately 50% efficiency and the maximum solar radiation is $1kW/m^2$, then for the four second period, the condenser is required to handle 5kJ i.e. 1.25kW.

This power input figure equates with the condenser's heat transfer calculation. It is probable that the wall temperature of the condenser will be slightly higher than the water temperature, depending on the water flows around the tube. To counteract this effect it is likely that the condenser will operate more efficiently than calculated because the steady state conditions assumed in the calculations will not be reached due to the condenser being in use only 40% of the time. When the condenser is not receiving vapour, it will still be cooling the liquid and vapour inside. This calculation has some inaccuracy associated with it because the flow around the condenser's tubing in not known.

6.1.2 Feed Pump

The feed pump transfers the vapour (which has been condensed in the condenser) from the bottom of the condenser to the solar collector. When transferring the fluid it must also pressurise it from the condenser to the collector pressure. The stroke of the diaphragm of the feed pump is 12mm and it has a surface area excluding the diaphragm of 2.0 x $10^{-3}m^2$. This unit will pump at least 24ml per stroke. This volume is greater than the volume of liquid condensate expected per stroke. The feed pump is attached via a sliding arrangement, to the pump's diaphragm. For the first 18 mm of travel of the pump's diaphragm, in either direction, the feed pump does not operate. In the final 12mm the feed pump is either pulled or pushed by a mechanical stop attached to the feed pump's diaphragm.

6.1.3 Pump Body

The flat surfaces of the pump body are made from 6mm mild steel plate. 3mm mild steel sheet is used for the cylindrical water jackets (the 3mm thickness was needed to ensure buckling does not occur at the low internal pressures). The base of the pump was machined from a 40mm mild steel plate.

The diaphragm was made from 1.6mm uncured Nitrile rubber, which was cured at 150°C for 20 minutes in its deformed position, in order to make the cups in the diaphragm permanent (A medium sized double thickness diaphragm is shown in Fig. 32). The centre of the pump was guided in its vertical motion to ensure that the diaphragms were not over-stretched or pinched by uneven motion of the central unit. The guides were machined from Ultra High Molecular Weight Poly Ethylene (UHMWPE). The lower guides rub on the outside of the inner cylinder, the top guides rub on the inside of the outer cylinder (one of the top guides can be seen at the top right of Fig. 31). Incorporated in the mount for the lower guides is a lip, which the diaphragm unit hits in order to stop it from travelling higher than the 30mm and damaging the diaphragms, valve or feed pump.

6.1.4 Water Piping and Valves

The pipes transferring the water into and out of the pump are standard 40mm PVC waste pipe with 40mm PVC fittings. The two non-return valves attached to the PVC piping are commercially available 1½" brass bodied IVR Art 90 valves.

6.1.5 The Pentane Side

All the pipes on the n-pentane side are copper, being ½" for the liquid return and ¾" for the pentane vapour supply. All of the fittings and valves are brass Cajon, Whitey or Swagelock products. The n-pentane itself was difficult to obtain in the less pure grades. The most readily available pentane was the 'Analar' grade (analytical chemical quality) at a price of $96.00 for 2.5 litres.

6.1.6 The Valve

The valve body was machined from a large block of mild steel, with most of the internal parts of the valve also being manufactured from mild steel. The springs were made from stainless steel because of its better retention of properties at the temperatures at which the pentane will work. The force exerted by the spring is 60N when fully compressed (20mm solid height) and has a free height of 30mm.

The seals in the pump's modified valve are of the V ring variety. The rings sealing lips take little force to seal and enable the use of a less-accurate machined valve than 'O' rings would require. The valve uses 1, 34mm V ring and 12, 8mm V rings.

6.2 TEST SET-UPS

In order to ensure easy set-up and accurate testing of the water-pump prototype, it was intended to put the prototype through three set-up stages. The first stage entailed simulation of the pentane pressures by compressed air, the second was the simulation of the solar collector by a pentane boiler and the final system, was the complete system with a real solar collector.

6.2.1 Pump Testing on Compressed Air

The compressed air stage involved the setting-up of the pump, ensuring that it operated as desired within its operating pressure range. The ancillary equipment needed to test the pump supplied: high pressure air to substitute for the high pressure pentane vapour, lower than atmospheric pressure air to simulate the low pressure pentane in the condenser and lower than atmospheric pressure water to simulate the several metre suction head encountered in a well. The auxiliary equipment consisted of reservoirs, regulators, pressure gauges, venturi suction units and relief valves, as shown in Figure 33.

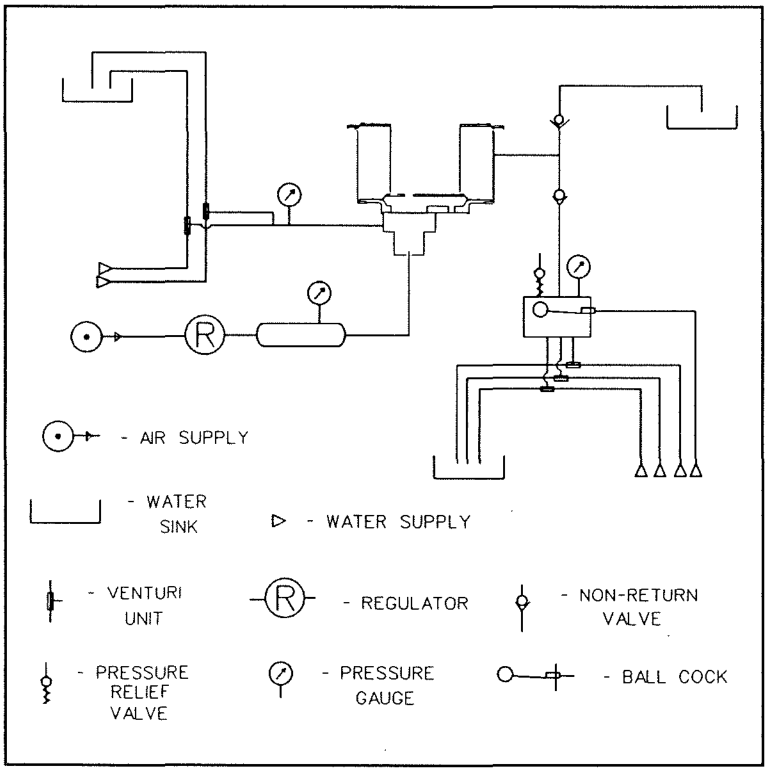

Figure 33 Air Simulation Schematic

The low pressure water was provided by using two venturis to lower the air pressure in a reservoir which was half full of air and half full of water. The water extracted from this reservoir by the water pump was replaced by two ball cocks. The pressure within the reservoir was able to be adjusted from atmospheric pressure down to 25kPa absolute by an adjustable relief valve. The pressure could be read from a pressure gauge attached to the venturi devices. The low pressure air to simulate the condenser was provided by three venturi suction units. The pressure was able to be adjusted from atmospheric pressure down to 5 kPa absolute by an adjustable relief valve. The pressure could be read from an attached pressure gauge. The high pressure air was supplied by two regulators and could be adjusted from 101kPa to 1,500kPa absolute. The pressure of the high pressure air was also able to be read from an attached pressure gauge.

The delivery head for the water was not simulated, as real piping with sealable exits at 0.5, 1 and 1.5 m were used. The prototype with PVC piping for the water and plastic hose for the air is shown in Fig. 34.

The intention of using air to simulate pentane pressures was to ensure quick and easy adaptation of the valve, with no need for purging the system with pentane after every adjustment to the valve. The easy disassembly proved to be very useful in the initial instance because problems occurred in getting the valve to operate as desired.

6.2.1.1 Problems with the pump's operation on air pressure

The major problem with this operating scenario related to the valve's failure to seal. The basis of the valve's operation was an over-centre type set-up in which either the valve's inlet or its outlet is completely open to the vapour chamber. In order to move the valve from one position to another the operating principle was that: the valve is held in position by a pressure difference across the seated area of the valve. As the pump's diaphragm pushes on the valve spring and hits the end stop, the seal starts to leak. This leak should change the pressure balance across the sealed area enabling the compressed spring to take over, to push the sealed surface away and hold the valve in its new position against the other surface to be sealed.

Figure 34. Pump Set-up on Compressed Air

Figure 35. Valve Internals, Showing V Seals

In the actual unit, the diaphragm pressed on the spring and reached the stop, the sealed area leaked slightly but it did not change the pressure balance across itself enough to allow the spring to push the valve to its other position. This problem was compounded by the fact that the seal was difficult to achieve even on an accurately machined surface. The problem was that the rubber O rings (which were used to seal the faces) were too stiff to take up any misalignment or distortion of the sealing surface.

Both of the solutions looked at to overcome the sealing problems in the valve required physical changes to be made to the valve body. The first solution requires V rings to replace the O rings. The V ring solution would allow for the sealing surface to be less accurately machined and would solve the problem of leakage due to misalignment. V rings would also have less problems due to spring forces. The spring problem would be reduced because, as the sealing surface starts lifting off from the V-seal, the seal will get to a point where the V membrane fails to seal because it reaches geometric constraints (rather than leaking). At this point the spring force would dominate, causing the valve to change position. The only foreseeable problem with this set-up is the life of the V-seals, as they will experience moderate deformations which they may not have been designed for. The second possible alternative to get around the sealing and spring problems is to use a totally different valve set-up. The current set-up uses the pressure across the valve to hold the valve in position. This could be modified so that the pressures of the working fluids do not influence the valve's performance. This valve would operate better under extreme pumping conditions. The most likely form of this valve would be a set of detente balls on a shaft. The end of the shaft has inlet and outlet holes drilled radially through it. The shaft would be driven by the pump diaphragm's displacement via a pair of springs as in the current valve set-up, see Fig. 36.

The configuration in Fig. 36 enables the valve to be set up to run with a certain set of springs. The valve will then operate for any water and pentane condition, because these pressures will have no effect on the valve's motion. The motion is not effected because all of the sealing surfaces are perpendicular to the direction of motion.

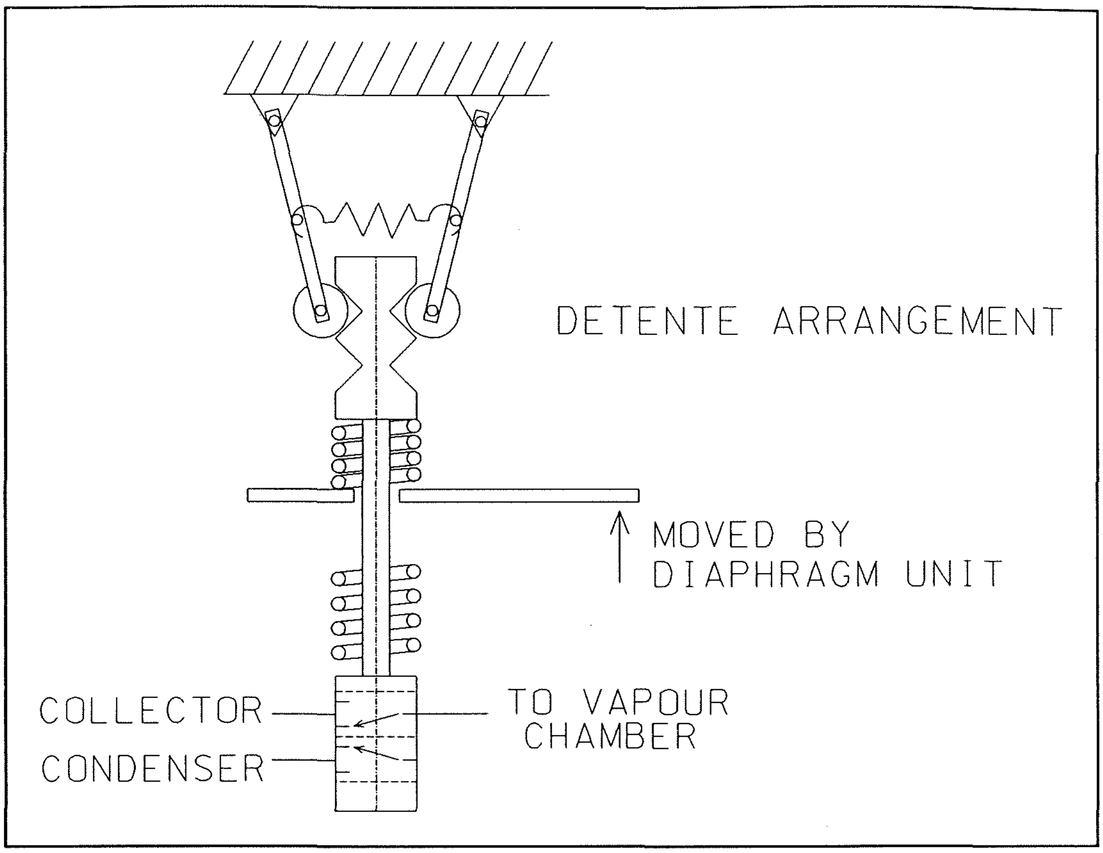

Figure 36. Ideal Valve Arrangement

The first alternative is the one which has been developed (the 12 V rings can be seen in Fig. 35), because the latter involves a total rework of the valving. The well machined surface for the second alternative valve does not fit the manufacturing ideals expressed in Chapter 3. The second set-up is perhaps superfluous to requirement because once a water pump is made and installed for a certain situation, conditions should not change drastically.

The valve with V rings sealed better but needed the high pressure air route from the start to the end of the valve to be enlarged to overcome undesirable pressures which built up in the more constricted passages of the valve. The flow of the gases through the restricted passages created a pressure which stopped the valve from reaching its final sealing position.

6.2.1.2 Pump output on compressed air

The output results for a pump operating on compressed air are expressed in Table 8. The cycle time was excessive because the venturis which simulated the condenser did not have the same pressure pull down capacity as the condenser, as a result 90% of the cycle time was spent in the condensing mode of operation. The low power of the suction devices made the pressure in the vapour chamber during the water delivery part of the cycle equal only to that which was necessary to expel the water with minimal acceleration or pipe losses. The high pressure supply air which simulated the pentane dropped from the supply pressure of 270 kPa down to 215kPa absolute when it was supplying compressed air to the vapour chamber.

Table 8. Compressed Air Results

Delivery head m	Suction head m	Total head m	Cycle time seconds	Water quantity litres	Collector pressure kPa (Air)	Condenser pressure kPa (Air)
0.5	1.3	1.8	11.2	2.365	270	93
1.0	1.3	2.3	12.3	2.355	270	86
1.5	1.3	2.8	14.6	2.34	270	80
0.5	4.1	4.6	12.2	2.185	270	93
0.5	6.4	6.9	13.3	1.945	270	93
0.5*	7.4	7.9	25.6	1.29	270	93
1.5	6.4	7.9	16.7	1.92	270	93

* The valve failed to operate properly at this water setting.

The lengthening of cycle time for increased delivery head is because more pressure is being used to do work (increasing the waters potential energy) instead of accelerating the water into the pump. The quantity of water delivered per stroke fell with increased pumping head. This effect was less obvious for the changes in delivery head and it was noted that the diaphragm remained bowed inwards for this part of the cycle, despite the fact that the diaphragm had a greater pressure inside it. The increase in suction head had a larger effect on the diaphragm and ballooning was evident by the fall in water output quantity, (the ballooning can be seen in Fig's. 37 and 38, where 37 has no pressure across the diaphragm and 38 has 250 kPa across the diaphragm).

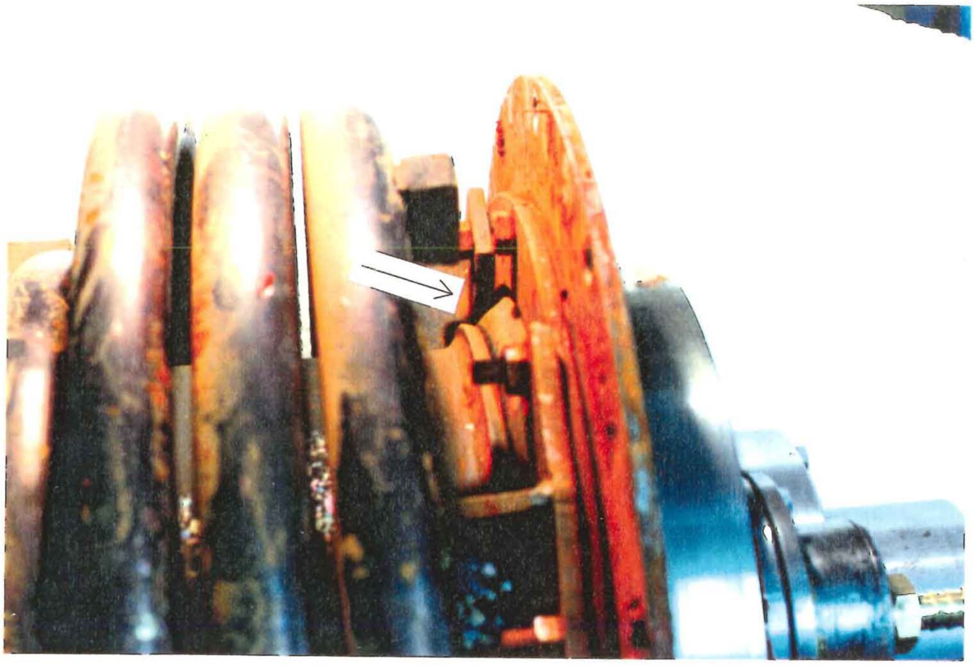

Figure 37 Unstressed Rubber Diaphragm

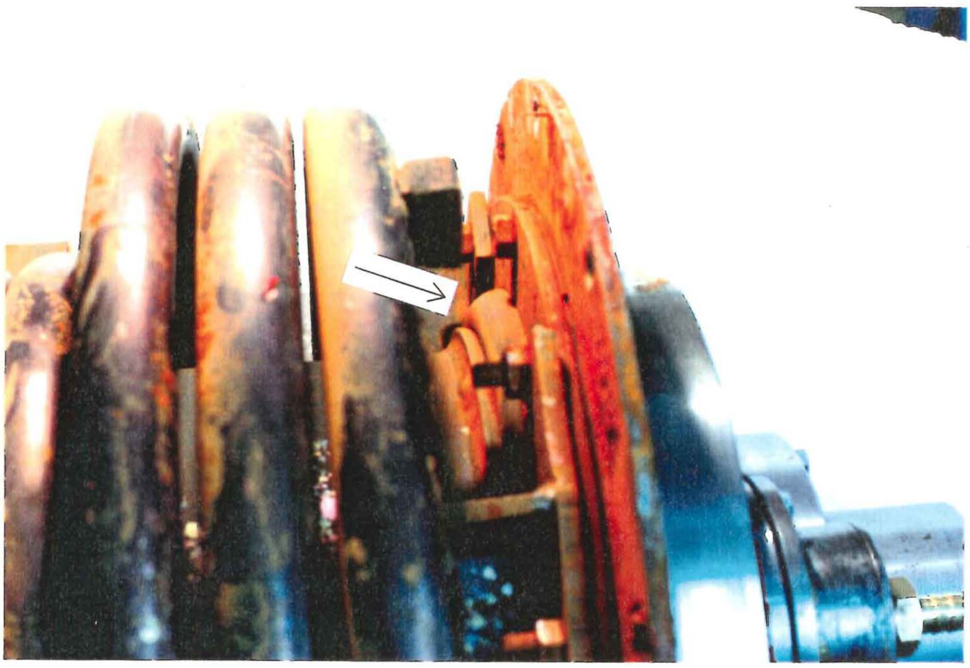

Figure 38. Ballooning Rubber Diaphragm

It is possible that any air trapped in the water chamber would be detrimental to the pump's performance. The trapped air's effects may be felt mostly at the greater suction heads. The valve failed to operate properly at the 7.4m suction intake, the symptoms being a leak of air from the simulated collector to the simulated condenser for a part of the cycle. This leak was probably due to the pump body not fully reaching the end stops of the valve during the intake part of the cycle. Therefore the valve was leaking a little but did not have enough force applied to it to fully break the seal of the V rings.

6.2.2 Pump Testing with Pentane (simulated collector)

The second major set-up was used to ensure accuracy of test results by gaining an accurate knowledge of the power input into the pentane boiler. A schematic of the set-up is shown in Fig. 39.

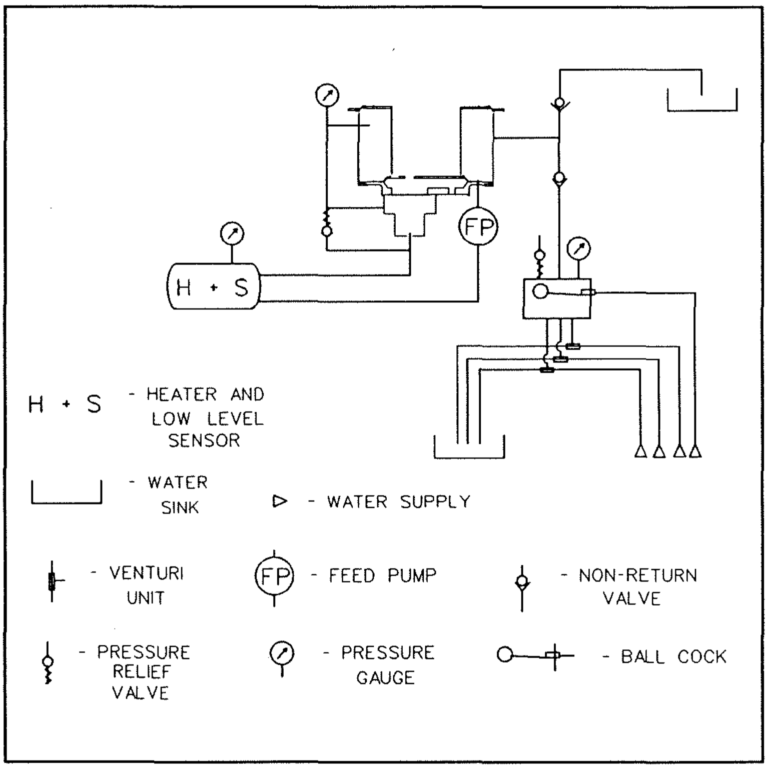

Figure 39. Pentane Boiler Schematic

This set-up does not need the compressed and decompressed air to simulate the pentane's pressure as it replaces the air with pentane. The pentane is condensed (in the condenser) within the pump, but the vapour is not generated in a solar collector. An accurate measure of the power into and out of a solar collector during operation would be difficult to obtain. A more accurate way to achieve a reliable reading is to replace the collector with an electric vapour generator. The vapour generator consists of 2, 1 kW hot water heater elements. In order to ensure the elements are never uncovered by the pentane, an optical level sensor and cut out relay are included in the vapour generator's power supply.

The amount of power into the vapour generator is able to be varied by using a number of heating elements in combination with a Variac. The temperatures and pressures of the vapour were able to be ascertained through the use of a pressure gauge attached to the vapour generator and condenser.

It was in this set-up that the pump's operating efficiency was able to be found. The test set-up with the pentane boiler is seen in Fig 40. The lower level of the table which the pump sits on has (from the rear): the low pressure water reservoir, feed-pump (partially obscured), pentane boiler and Variac.

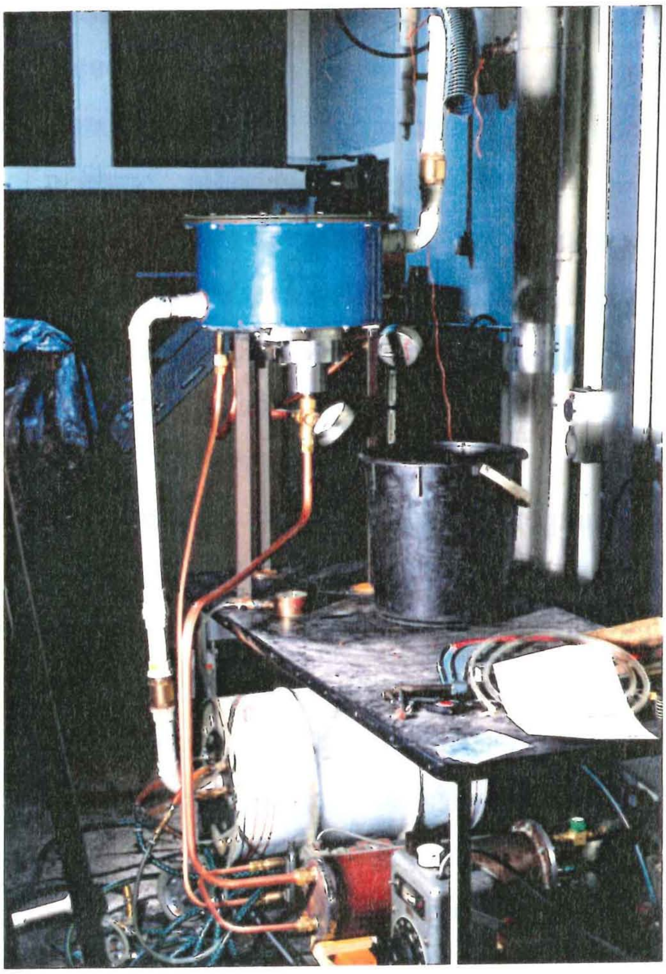

Figure 40. Pump using Pentane with Simulated Collector

The operation of the pump on n-pentane was only partially successful due to a leak from the vapour chamber into the water chamber. This leak allowed the high pressure vapour to condense into the water, reducing the quantity of pentane in the boiler. After 15 minutes of operation, the level in the boiler had dropped sufficiently for the low level sensor to switch the vapour generating elements off. The leak made the condenser less powerful i.e. low pressure was not able to be achieved, despite the fact that the condenser was below 20°C. This higher than desired condenser pressure indicates that gas other than pentane was getting into the condenser through the vapour chamber.

The partial results from this section of testing were: in a 24 second cycle time with 1.3 kW power input to the vapour generators, 2.1 litres of water were pumped from 1.3 metres below the pump to 0.5 metres above the pump. This gives a crude efficiency of 0.1%. It is completely possible for this efficiency to be increased because during the period in which pumping was taking place:

- the vapour generator and all the copper tubing was not lagged,
- a fan was blowing cold air (13°C) over the whole apparatus to ensure no build up of pentane vapour took place,
- there is a possibility that the pump was not in fully steady state operation due to the vapour chamber's walls being below the vapour's temperature,
- pentane was escaping from the leak in the pump, boosting the consumption of energy from the boiler.

The pressures on the pressure gauges during the pumping cycle were: Collector pressure built up to 255 kPa before the valve opened and dropped down to 195 kPa when the vapour chamber took vapour from the boiler. (Note, this is very similar to the air supply pressures for the same situation in the previous tests using air instead of pentane). The condenser pressure was 95 kPa (absolute) during the water delivery phase and did not change significantly when not in use.

The pressures in the water chamber were read from a pressure gauge and gave a minimum reading of 50 kPa for water intake and a maximum of just over 100 kPa for water delivery. These pressures indicate that sufficient pressure is available to accelerate the water into the pump, but little is available to deliver it from the pump (the same problem was encountered with the air set-up). Considering that the supply pentane and condensing pentane conditions were very similar to the air's simulation conditions, the operating results could be transferred to the pentane simulation, i.e. the cycle time increase for different heads and the change in water output for different heads could be transferred to the pentane simulation, based on the pentane result presented above, see Table 9.

Table 9.Extrapolated Pentane Results

Delivery head (m)	Suction head (m)	Total head (m)	Cycle time (sec)	Water quantity (litres)	Power input (kW)	η (%)
0.5	1.3	1.8	24	2.1	1.3	0.12
1	1.3	2.3	25	2.1	1.3	0.15
1.5	1.3	2.8	28	2.0	1.3	0.15
0.5	4.1	4.6	26.1	1.9	1.3	0.26
0.5	6.4	6.9	28.5	1.7	1.3	0.32
1.5	6.4	7.9	32.5	1.6	1.3	0.30

The table indicates that the greatest efficiency is found where the pump is doing its most useful work (i.e. water lifting). For lower pumping heads the useful work which is available to be done is absorbed by parasitic factors such as fluid acceleration and pipe and fitting losses.

The valve worked well from a cold start-up with no problems due to leakage or faulty operation evident. These results indicate that the valve is reliable in this mode of operation, i.e. under start-up or run-down conditions.

It should be noted (from the pressure gauge readings) that there is a drop in delivery pentane pressure when the vapour flows through the pipe work into the vapour chamber. This pressure drop warrants inclusion in the computer simulation because it has been shown to be of a reasonable magnitude.

6.2.3 Solar Collector Set-up

The set-up using a solar collector was 'the' solar powered water pump, but the testing did not progress far enough to enable the pump to be attached to the solar collector. The collector set-up would only have been useful to confirm the simulation program's results for cloudy periods, i.e. to what temperature the collector will fall and how much down time is predicted for the whole system when a cloudy period is encountered.

6.3 COMPARISON WITH COMPUTER MODEL

A comparison of the water pump's performance with that predicted by the computer program, was not able to be undertaken fully due to the obvious discrepancy of the condenser's pressure in the real pump unit being excessive due to the leakage of foreign gasses into the condenser. It was possible however to alter the condenser pressure value attained by the simulation program to try and produce the same conditions which were being experienced by the real pump. The simulation of the pump for one cycle with a fully functional condenser produced predictions of a seven second cycle time (2 second water intake stroke) and 2.3 litres of water being pumped. The restricting of the condenser pressure value was done by setting an upper limit to the pressure which was available to accelerate the water out of the pump. Exclusive of all losses, an accelerating force for the water delivery stroke of 200 N, produced the same cycle time as the real pump operating in the simulated collector set-up. This comparison is of dubious value due to the thermal losses (mentioned in the previous section) experienced by the pump in its early stages of prototype testing. The water output of 2.3 litres and water intake time of approximately two seconds were in good agreement with the real pump, but until the pump operates as a fully functional unit in steady state conditions with the same degree of lagging of the high temperature areas as is assumed in the program, the comparison with the program's output will be of dubious value.

7 FINANCIAL AND FUNCTIONAL VIABILITY

Developing countries needs lie in the field of intermediate technology devices which are cheap to produce within the country concerned. The units should also be very reliable in operation and effective in their designated role.

7.1 THE FINAL SOLAR THERMAL PUMP'S COSTS

The true cost of a water pump needs to include in-service costs as well as the initial material and manufacturing costs. In-service costs may be maintenance, parts, fuel, contribution towards a new replacement pump and depreciation on the initial investment. The benefit of water pumps on the local scale is the volume of water produced when needed. Benefits of locally produced water pumps on the national scale may be those from lower foreign produce purchases, some employment in the manufacturing processes, skills and experience gained. Less reliance on overseas technology, expertise and personnel would be advantageous for the developing country, allowing for reliable delivery of the pump and replacement parts. A reduction in dependence on foreign countries would ensure the system is internally regulated rather than externally dictated.

7.1.1 The Pump Unit's Initial Cost

To cost the pump unit it was desired to use the maximum number of commercially available components which could be costed at the retail price (inclusive of GST).

It is assumed that the pump's manufacturer would buy from the same wholesaler or producer as the retailer who costed the components. Therefore the profit made by the retailer is adopted by the pump manufacturer. If components are purchased from a non-retail source it is expected that a discount for regular and large customers would be available. In this study these components had no added value except for development and assembly costs. This is justified in that, for the sake of this costing, the mark-up and overheads associated with each of the components (by the retailer) was adopted by this study as the mark-up and associated overheads for these products.

Labour associated with the production or machining of parts for the pump is taken at $40.00/hour. This includes the cost associated with the machines being used for the machining, and overheads of the premises where the pump is being produced. The costs of packaging and recovery of development costs take the form of 2% added to the final manufacturing, labour and parts cost. Likewise the final mark-up of 10% is placed on the labour costs. 12.5% for GST is added to the final figure.

The individual components were priced as follows:

		Retail	Labour
Diaphragms*:	large diameter diaphragm	40	
	medium diameter diaphragm	30	
	feed pump diaphragm	5	

*Price for commercially available cupped composite diaphragm for similar size pumps. Actual nitrile cost is $6 for the large diaphragm.

Mounting legs: for sitting in concrete or attachment to a frame.

		Retail	Labour
	angle iron x 3m	35	
	miscellaneous mounting flanges	10	
1 hour labour for fabrication and machining			40

Pump body: either pressed stainless steel or cast aluminium

		Retail	Labour
	material and 2 hours labour	170	80
	diaphragm holding rings	113	60
	60, M8 nuts and bolts	24	

		Retail	Labour
Pentane*: 2.5l		96	

*Analar grade (high quality analytical chemical grade). It is probable that a supplier of a cheaper, less refined product is able to be found.

Solar collector: 2.9m^2 made from copper tubes woven in aluminium sheet with transparent polypropylene box section cover. 1475

It is possible for this item to be reduced in price as other collector varieties can cost as low as 290/m^2 i.e. $841 for 2.9m^2

Condenser: copper coil and attachments			
	copper tube 1" x 5m	39	
	fittings x 6	18	
	condenser housing	63	
1.5 hours labour for fabrication			60
Water side:	2, IVR Art 90/104 non-return valves	98	
	40mm PVC waste pipe 8.5m	38	
	elbows	6	
	fittings (assemble on site)	22	
Liquid and vapour lines:			
	copper tube ½" x 4m	17	
	copper tube ¾" x 4m	23	
	check valve x 2	60	
	copper tube adapters	30	
	½" T	20	
	1 hour assembly		40
Guides:	channel x 1m	20	
	UHMWPE	10	
	labour for welding and fabrication		80
Feed pump:	body	14	
	bolts	3	
	1 hour labour		40
Main valve:	body material	50	
	bolts	6	

	V rings	18	
	miscellaneous washers etc.	20	
	spring	7	
	3 hours labour		120
Assembly:	5 hours labour		200
TOTAL		**2580**	**720**

Totalling labour = \$720 x 1.02 = 734

Totalling parts = \$2,580 x 1.02 = 2,632

Mark-up + GST on (labour x 1.1) = 807 x 1.125 = 908

Total Cost = 908 + 2,632 ≈ \$3,500

It is suspected that the solar collector is overpriced, this is probably due to it being an infrequently made item and having many wholesalers or intermediaries mark-ups added to the final production costs. The material costs are as follows:

Al sheet 1.2 x 2.4 m	\$87
Cu tube ½"	\$102
Clear, twin walled 'Clearon'	\$85
Cu header pipes	\$37
Insulation, (rear)	\$40
Case & Backing 1.2 x 2.4 m	\$70
Painting	\$50
≃	\$470

Thus ≃ \$1000 Labour, overheads and profit!!

There is scope for reductions in the water pump's capital costs with both medium and large scale manufacture. There is also the possibility of this unit being made in a developing country, where the labour costs are less, e.g. Halcrow and Partners

(1983b) suggests 25 to 42¢/hour for field workers and Kenna et al. (1985) suggests \$1.70/day. A skilled metal worker would command a better wage than those suggested, but (including overheads and machine costs) the total labour figure may be lowered by up to 50%. This reduction in labour cost and the use of a less costly solar collector, could see a well manufactured unit at the price of \$2500. Mass manufacture may see the price drop even further with bulk buying and better jigs both reducing the materials and labour costs.

7.2 VIABILITY COMPARED WITH OTHER WATER PUMPING DEVICES

The viability of low powered systems (20W), will be substantially different from larger scale units (>100W) because for the small scale units, the prime movers are relatively more expensive. For example, a small turbine, while being very efficient, would cost several times more than a piston or diaphragm pump. The solar collectors also face the same problem, i.e. a tracking concentrating collector would be prohibitively expensive and thus not justified when the final power output is only 20W. The efficiency of size is against the small higher technology solutions because they need the same components as a larger unit but have a lower power output. There are designs for simple sun tracking mechanisms (Radajewski, 1987) but they would only add to the complexity of the pump and provide another area where problems could occur.

The several units which the solar-thermal pump is compared to, range from units which would not be compatible with the Village Level Operation and Maintenance scheme (VLOM) to some units which may fit into a VLOM scheme with suitable modification, see Table 10.

Table 10. Alternative Pumps and Performance

Unit	**Capital cost inc. GST**	**Pump type**	**Flow at 8m head**
Windmill [7]	$ 2648	51mm glandless	400 l/hour [1]
P.V. 1 panel [3]	$ 4331	+ve displacement	204 l/hour [2]
2 panels [3]	$ 5546	"	409 l/hour [2]
Petrol diaphragm[6]	$ 2364	diaphragm	1280[4] l/hour
Petrol centrifugal[6]	$ 1176	centrifugal	27000 l/hour
Solar thermal [7]	$ 3300	double diaphragm	1080[5] l/hour
Hand pump [7]	$ 450	displacer cylinder	1200 l/hour

[1] with a 15 to 20 km/hr wind speed
[2] during peak sun hours
[3] 48 W panel
[4] includes capacity reduction factors from manufacturer
[5] for design radiation
[6] $164 added to manufacturers price for below ground components
[7] possible compatibility with VLOM scheme

All of these units, with the exception of the hand pump, need to have a storage tank of up to one day's water output. The storage is necessary for the low powered pumps (e.g. windmill and solar) because they would otherwise pump water when no water holding device is at the delivery point. For the petrol-centrifugal pump, storage is needed because the flow rate is otherwise too great to allow filling of typical water receptacles. Because all the units need this storage, it is not necessary to go into the area further and its absence will give a less damped indication of the best or most viable pump, i.e. the same storage system will give the same costs to all units and thus is not a variable which needs to be controlled for.

7.2.1 The Options

The long term costs of the pumps are now to be considered on an individual basis.

7.2.1.1 Hand-pumps

The hand-pumps tested by CATR (1984) generally failed after 4000 hours of use (equates to two years). At this point in the pump's life, total overhauls were necessary due to general wear of the bearings, piston, cylinder wall and seals. This wear and failure is caused by two years worth of rubbing contact (e.g. in the piston) and two years wear of mechanically powered bearing elements. Such replacement every two years could amount to ¼ of the hand-pump's initial cost. The replacement would need to be carried out by a person who was better trained than the village caretaker/maintainer. The hand-pump also requires an operator. This task is typically carried out by the water gatherers (women and children). Whether person power can be included in the financial equation is a matter for some debate as it is not one person spending 5½ hours pumping, but a multitude of people each filling their own containers. The fact is, that they would spend the same amount of time collecting water whether they pump the water or wait for the sun to pump it at the same rate for them. For this study, the labour is considered more intensive when pumping by hand, than by merely filling the container and thus the lower rate of 25¢/hour will be used as the labour rate.

The hand-pump should be considered as the base condition because hand-pumps are currently in use and any alternative would have to be better than the hand-pump to be accepted as a successor.

7.2.1.2 Photo Voltaic pump - 1 and 2 panels

The long term performance of a solar photo voltaic (P.V.) pump has been documented by Maheshwari (1985). This Indian study suggests that the P.V. modules of the unit tested had a limited life, operating for an average of 21 months before failure. The reasons for module failure were open circuit (60%), thermal stress (18%), and various other defects. Burgess et al. (1985) assumes a five year array life; this study assumes a three year P.V. array life. The solar P.V. pump used, requires an overhaul of brushes after approximately 6,000 hours, but as has been noted by Burgess et al. (1985), such units use water lubricated bearings. Frequently, an automatic device like the P.V. pump has been started up when the well is dry or has been damaged due to operation with a low water table. The overhaul costs are assumed to be one tenth of the pump unit's initial capital cost. It must be assumed that a supply of bearings and brushes for the unit is readily available. The replacement of brushes would need an experienced person, as the unit would not be compatible with a VLOM scheme. There should be little labour time associated with the daily operation of the P.V. water pump. It is possible to move the P.V. array to face the sun morning and afternoon, with a resultant gain of 30% more water output (Maheshwari, 1985).

7.2.1.3 Windmill

The windmill driven units have the same type of pump unit as the hand pump and will incur the same costs at approximately the same frequency as the hand pumps. There will be an extra cost incurred with the maintenance of bearings in the windmill and corrosion prevention of the windmill's support structure. The frequency of bearing replacement is approximately once every two years. This job needs an experienced overhaul person. There is no need for any daily attention to the windmill, although a monthly greasing would be of benefit.

7.2.1.4 Petrol powered units

These units require the greatest amount of care in operation, yet give the greatest water output for the labour input. The daily starting and checking of oil and fuel and possible daily purging of the water side, is only a little labour intensive and the pumping results are large. The unit needs to run for less than an hour every day to supply the desired water (centrifugal version) or for a six hour shift for the diaphragm version. Regular replacement of oil, petrol, air filter and oil filter are necessary, with an operating life of 3500 hours before overhaul (of rings and bearings). The overhaul would probably cost one third of the initial capital cost. Fuel is used at the rate of 300 grams for each kilowatt of engine power per hour of operation (Blackmore et al. 1978).

7.2.1.5 Solar thermal pump

This pump unit should need no daily attention for its satisfactory operation unless a non-return valve leaks, then it would require daily purging of the water side. An annual replacement of the diaphragm will be necessary to ensure reliable operation. It is probable that the valve seals and pentane level will also need to be checked at this interval. These operations will need to be carried out by a person with more experience than the village caretaker.

The above is summarised in Table 11

Table 11. Alternative's Maintenance Requirements

Unit	Maintenance period	Parts to be replaced	Replaced by whom	Daily requirements
Windmill	2 yearly	bearings,pump	o/haul person	none
P.V. 1 panel	3 monthly	collector arrays	caretaker	none
P.V. 2 panels	3 yearly	brushes	o/haul person	none
Petrol diaphragm	monthly	oil,filters,etc	o/haul person	starting + fuel
Petrol centrifugal	4 monthly	oil,filters,etc	o/haul person	starting, fuel + bleeding
Solar thermal	yearly	diaphragm	o/haul person	none
Hand pump	2 yearly	pump cylinder	caretaker	continuous

7.2.2 Final Viability at Different Discount Rates

It has been suggested by Burgess et al. (1985) that many of the field trials of water-pumps have failed because of the working costs of a pump not being matched by farmers who may put a greater emphasis on the purchase of fuel for transport to market rather than water pumping. It was suggested that many pumps come with low or no initial capital cost because aid agencies supply the pump. The finance for running expenses is then left to the individual farm owner. For this reason the cost of daily running expenses was given a higher rating than initial costs. This cost rating led to wind and solar power having an advantageous position due to low running costs. The study gave a ranking according to different discount rates but the final cost figures were of no real relevance due to multiplication by a theoretical discount rate. The stance taken by this study will be to discount the capital cost (as happens in reality) but also to include the cost of investing in a replacement pump when the functional life of the current pump has expired. This perspective gives a better overview of the costs that might be faced in the real world, rather than just a ranking dependent on discount factor.

For the pump considered above, the costs are portrayed in Table 12.

The three bottom lines represent the current cost of all the options, given a 5% yearly discount rate, no residual capital value and a 20 year period of operation.

Table 12. Financial Summary

Pump type	Windmill	Photovoltaic 1 panel	2 panel	Petrol diaphragm
Capital cost	2648	4331	5546	2364
Hours to get 6m^3	15	20	10	4.7
Recurrent costs	299	1005	1110	1691
Pumping labour	0	0	0	0
Yearly recurrent cost	300	1000	1100	1600
Assumed life (years)	20	20	20	6.6
Capital cost	2648	4331	5546	2364
Maintenance	3726	12524	13833	2991
Replacement	923	1632	2090	890+710+481
Labour	0	0	0	0
Energy	0	0	0	18081
Total $	7297	18490	21470	25520
50% of 1st pump free $	5973	16313	18697	24338
Free 1st pump $	4649	14159	15924	23156

Table 12.cont Financial Summary

Pump type	Petrol centrifugal	Solar thermal	Hand pump 0c/h 25c/h 50c/h		
Capital cost	1176	3300	450	450	450
Hours to get $6m^3$	0.22	5.5	5	5	5
Recurrent costs	214	205	76	76	76
Pumping labour	73	0	0	456	913
Yearly recurrent cost	280	205	76	530	990
Assumed life (years)	20	20	20	20	20
Capital cost	1176	3300	450	450	450
Maintenance	747	2554	947	947	947
Replacement	443	1243	170	170	170
Labour	909	0	0	5682	11377
Energy	1919	0	0	0	0
Total $	5194	7097	1567	7249	12944
50% of 1st pump free $	4606	5447	1342	7024	12719
Free 1st pump $	4018	3797	1117	6799	12494

It can be seen that the two P.V. pumps and the petrol driven diaphragm pump are totally unable to be considered, irrespective of the way in which the initial capital costs are reduced. The reason for this is: (for the P.V. systems) the initial cost and replacement cost of the P.V. arrays on failure, are excessive. The P.V. also lacked performance, needing a daily operating period of 20 hours (one P.V. array) to pump the required amount of water. On a good day 8 hours is the maximum length of operation that can be expected from these pumps. The above considerations lead to the conclusion that the P.V. unit is underpowered and overpriced. This is probably due to it being a novelty item in New Zealand. The second pump which was not acceptable was the petrol-driven diaphragm pump. It uses a 1.6 kW motor to drive a diaphragm through a 'V'-belt and a reduction gearbox. The major cost of this unit is the large fuel bill. The low water output indicates that the actual means of pumping is most inefficient when compared to, say a centrifugal pump. The actual capital cost of this unit may be excessive, due to it being an Australian unit, imported into New Zealand.

There are many underlying assumptions throughout the economic viability study; the first is that pumps and components will cost the same in both New Zealand and the developing countries. This assumption is justified by the scarcity of the pumps in the New Zealand market, as is the case in developing countries. It is assumed that the interest earned on money invested is 5% higher than inflation. It is also assumed that most of the units have a 20 year life, at the end of which there is no residual capital value. The only exception to this is the diaphragm petrol pump which is operated for a long period daily and thus only has a 6.6 year life expectancy.

The estimation of the recurrent costs is open to a fair degree of variability.

The use of 0, 25 and 50 cent labour charges for the hand-pump, produce very large changes to the hand-pump's final life cycle cost. Note that in 1985 the gross national product for India was $433 per capita which, when calculated at present worth for 20 years at 5%, equates to $5,200 which is very similar to the labour charge for the hand pump at 25¢/hour. Halcrow and Partners (1983a) assumes 33¢/hour for irrigation hand pumping and zero for village pumping. The reason for this has been looked into by Burgess et al. (1985) where work and essential labour is discussed with its relevance to hand pumping.

The four other pumps give an interesting indication of the possible economies of various forms of water pumping. The solar and wind pumps work out to have similar life cycle costs, but as the percentage capital cost of the pump unit is reduced, the solar thermal pump becomes more economic due to lower running costs. For the centrifugal petrol pump, (which is the best financial alternative with no reduction of initial capital value) as the initial capital cost is reduced to zero, it becomes less economic when compared to the solar pump due to its having larger repetitive costs. Depending upon the scenario and final use of the hand pump, it varied from the cheapest to the most expensive of the alternatives. This is discussed further in Section 7.4.

7.3 SUMMARY OF OTHER VIABILITY STUDIES

The most comprehensive study of water pumping alternatives was carried out by Halcrow and Partners (1983b). The results for irrigation in this study priced, from most expensive to least expensive: human, solar P.V., animal, wind and then diesel power as being the most economic. The solar and wind systems were noted for their relatively high capital costs. The final prediction (with 70% future price reduction for P.V. arrays) is for P.V. units to become competitive with diesel. For the village water pumping scenario the human powered pump fits between wind and diesel. For smaller units this study suggests that diesel units lose their viability, the order being solar, diesel, human with wind being the cheapest alternative. The study of Halcrow and Partners (1983b) would have priced the petrol, wind, hand-pumps and solar P.V. as $4,800, $1,000, $250 and $1,432 respectively (Note, no solar thermal unit was included in the Halcrow and Partners (1983b) study). This costing would give a totally different result from that achieved in the previous section due to the difference in initial capital cost estimates.

A study of the current costs of Rankine cycle, solar powered engines (Barber, 1977) concludes that for the large units studied (in the order of 50 kW) the total cost of the solar collectors is generally considered to be twice that of the Rankine engine. The study suggests that the most desirable factor is cheaper solar collectors. The final conclusion of Barber is that the solar-thermal alternative is not financially feasible except for areas where no other alternative (such as wind or fossil fuel) is available. Other studies have come to similar conclusions when discussing uses for small systems in developing countries.

For systems delivering water from depths of 90 m at the rate of 15 m^3 per day Onyeonwu (1978) concentrates the appraisal on land areas with no other readily available power and concludes (after a small economic calculation) that a P.V. pumping system would be almost feasible and suggests field trials take place. It also suggests local manufacture of low technology items such as flat plate collectors and solar drivers be undertaken.

The general situation in which a small solar pump would be of use was looked into by Howes (1984). Focusing on the situation of small farms in Pakistan, Howes' project supervised field testing of small scale P.V. pumps. Many areas of Pakistan were rainwater or canal fed, therefore did not need solar pumping. Two hectares was considered to be the minimum land holding for subsistence living for a family, but in more populated areas the average land holding falls well below this. These two hectare plots account for 58% of the farms in Pakistan, but only 8% of the total farm land. The use of animal powered Persian wheels is suited to irrigation of land areas of greater than 10 hectares. It is possible, however, for small farmers to cooperate and have one Persian wheel between several small subsistence farms. The introduction of new technology (such as tractors) for large farms enables on a large scale the working of otherwise uneconomic land, and causes the eviction of small tenants. It is possible that small solar pumps could make the tenants land more viable on a small scale, but due to the threat of eviction tenants are hesitant to invest. The comparison of costs in this study focused on 250 W P.V. pumps and compared them to persian wheels. The final conclusion was that P.V. sets for irrigation purposes were 12 to 30% more expensive than current systems. This report went on to suggest the need for smaller power units for the less than 2 ha land users. It also suggested that the allocation of subsidies by the government should be extended to cover these smaller solar water systems, as under the current system, only water pumps with moderate outputs qualify for government subsidies.

A brief look at wind powered systems by Burgess et al. (1985) suggested that in the Sudan and Africa in general, the average monthly wind speeds are lower in areas where irradiation is greater. There are vast areas, however, which have a mean wind speed of over 3.5 m/s and thus were quite viable for wind pumps.

7.4 SUMMARY OF VIABILITY & FINANCIAL PROBLEMS

The construction and testing of a water pump has involved trade-offs between many principles in order to achieve a pump which is realistically able to be manufactured and has an acceptable work output. The final nature of the trade-offs and extent to which the final unit compromises the initial ideals is now to be examined.

7.4.1 Physical Attributes

Pump materials and corrosion

The concept proving pump was designed for ease of inexpensive one-off manufacture and thus was not designed with corrosion as a major concern. For the final unit, the choice of materials for the water and pentane sides of the pump will be critical to the pump's life. The water side requires a material which does not corrode when in contact with the water or contact with incompatible metals within the water. The same material will need to be compatible with n-pentane, as incompatibility will not only cause corrosion but could give off gases on decomposition. These gases would create a partial pressure detrimental to the performance of the pentane vapour especially in the condensing stage. For construction, the two most likely materials for the pump body would be spun stainless steel or cast aluminium. Both materials have good resistance to pentane and water. Stainless steel, however, is vulnerable to crevice corrosion on the water side if not flushed out with water after use. It is possible for this crevice corrosion to occur under mud, sand etc. which has built up in inaccessible corners. This type of daily maintenance is to be avoided for the village pump's scenario. Aluminium seems to be a more feasible choice. It may be possible to use other materials combined with the water pipe-work made from galvanised pipe. This pipe would act as a sacrificial anode and would require regular replacement. A more suitable pipe system would be PVC piping, avoiding the pitting corrosion at the joints of the galvanised pipe. Other benefits include: no bi-metal corrosion between the brass foot valve and the pipe work, light pipe enabling easy insertion and extraction from the well and low cost in terms of both capital and maintenance costs. One factor which may be a problem, is that PVC is more prone to shattering or breakage when struck or deformed.

The bolts in the pump should be cathodic to the pump body and thus steel or stainless steel bolts would be acceptable with an aluminium body. The condenser in its prototype position may cause problems because of dissimilar metal corrosion. To avoid this corrosion problem, it is suggested that the condenser be removed from the pump body and be relocated in a less critical area. The relocation fits in well with other functional concerns relating to the pump's operation.

The ability of the pump to self prime, should it be necessary, is hindered in the current set-up by a large dead volume associated with the condenser being in the water chamber. In order to reduce this dead volume the placement of the condenser in a separate tank above the second water non-return valve would be ideal. This chamber does not need to hold much pressure, so a cheap plastic container would house the condenser without any corrosion or functional problems.

The pentane side's pipe-work should involve mainly copper or brass fittings. These fittings are compatible with the copper collector tubes and the stainless or aluminium pump body.

The valve body and feed pump may also be made from cast aluminium, mild steel or stainless steel with stainless steel or steel internals. The valve also needs some extensive modifications to its structure, to make it easier to produce and assemble. The final valve should have sealing surfaces which need low machining accuracy and use more readily available materials in its construction.

The diaphragm (which is in use in the concept proving pump) was made from uncured nitrile rubber. The diaphragm was formed into its cupped shape and then cured in an oven. It was not possible to include any fibre reinforcing in the rubber and as a consequence the rubber balloons to a small extent as can be seen in the photos in Fig. 38. The inclusion of a composite in the diaphragm would stop this ballooning effect to an extent while retaining the flexibility of the pure rubber. It would be possible to include a reinforcing material in the diaphragm if it were made on a medium scale rather than one-off production.

An improvement in the pump's performance would also be expected with this change because more water would be pumped for less working fluid usage. This occurs because the excess dead volume in the vapour chamber is reduced due to the non-ballooning of the diaphragm. The water chamber will take in more water if the diaphragm is not ballooning into it.

The life of the diaphragm has been estimated as one and a half years or approximately two million cycles (Mernagh, 1969). It is suggested that a yearly replacement of the diaphragms be undertaken to ensure continuous operation of the pump. There is no real alternative to this diaphragm arrangement but one way to extend the replacement period would be to use a larger volume pump which cycles in double the current unit's time. Thus the major wear factor for the diaphragm (deflection) will be reduced by a factor of 2, therefore the diaphragm would have twice the life.

The actual removal and replacement of the diaphragm will be a process that requires a clean environment because the vapour chamber is open to the atmosphere. When replacing the diaphragm, the seals and possibly the guides, in the valve will be readily accessible and should also be replaced.

The process of removing and installing the diaphragm and seals would need to be carried out by a person with a good mechanical knowledge. This person would also need a suction device to purge the system of unwanted air. In addition, they would need to top up the collector with pentane. This is a technical job and would warrant the two tier approach which has failed in previous trials. It is unfortunate that this process cannot be carried out by the village maintenance person. Because of the complexity of this automatic pumping system compared with a hand pump, perhaps it is unreasonable to expect it to fall into the realms of a simple hand-pump maintenance scheme. It may be possible for a unit which has no exotic working fluid (e.g. water, as in the vapour piston set-up) to be maintained by the village level maintenance person, but such a unit would involve a reduction in efficiency.

The means of assembly of the pump, involves the use of 24 bolts and nuts. These nuts and bolts must form a pressure seal to stop water leaking past them. This is not a very desirable set-up, but a more viable method is yet to be suggested. It may be possible to have the bolts solid mounted in the sealing rings, but as such it is impractical to manufacture.

The major changes which should be a feature of the next pump are:

* relocation of condenser
* valve's use of readily available materials for internal componentry, i.e bolts, nuts, washers.
* feed pump's outlet position to be raised to enable self priming
* lower dead volume in both the water and vapour chambers

7.4.2 Financial Attributes

The table of costs (Table 12) illustrates the relatively high initial cost of the solar thermal pump. Once purchased, it is possible for the thermal pump to hold its own financially except when compared to the no-labour-cost hand pump. This leads to two points for discussion.

A. How to reduce capital costs

B. The value of human labour in developing countries.

7.4.2.1 Capital cost reduction

As suggested earlier in the chapter, the capital cost of the solar-thermal pump can be reduced by mass manufacture or purchasing direct from the producer for the constituent parts. At best, the price of the thermal pump unit, excluding the solar collector, could be the same as that of the centrifugal petrol pump. Adding $1000 to the price would include a 3 m^2 solar collector. It is probable that the complete solar thermal unit will never cost less than the petrol pump and thus the benefits to make solar units viable must be in the daily running costs. There are possibly indirect ways of reducing the pump's cost to the local users. These include foreign aid, local government subsidies or incentives, and foreign government aid schemes.

* Foreign Aid: This typically takes the form of a one-off grant of money to the government for relief purposes or to an international aid group to distribute as it sees fit. If a pump were to be purchased with this type of money, the hand-pump, windmill and solar thermal units would be of greatest benefit to the recipients. Since the recipients have very little money, they would not be able to afford to run a unit with high daily running cost.

* Local Government Subsidies: These subsidies generally apply to large farmers in the form of tax incentives or reduced initial capital cost. The case in Pakistan as discussed by Howes (1984) is that the local government subsidises pumps with a power output of over 160 W. The scheme allows for a 10% down payment and seven annual payments, with a lower than market interest rate applied. A scheme such as this for small water pumps would still benefit those systems with low capital costs and high daily running costs.

* Foreign Government Programmes: These programmes are generally aimed at providing for one sector of the population. Such programmes provide an ongoing effort to improve that particular socio-economic group's life or livelihood. The pump unit that is most applicable for this situation depends on the emphasis that the foreign government places on the value of initial and repetitive costs. The New Zealand Department of External Relations and Trade provides money to the South Pacific. This money is aimed at improving women's life styles and improving sanitary water supplies at village level for the less affluent villages. The solar-thermal pump would fit these criteria. Aid from this department may possibly go to developers of systems which are to be field tested in the Pacific region.

7.4.2.2 Value of human labour in developing countries

The study of water pumping costs by Burgess et al. (1985) included a large section of study on the value of human labour. For the purpose of farming or cropping the extra time which would be available if an automatic pump were used would only provide financial benefit for fallow systems, grazing systems and arable irrigation farming. The extra time made free by not needing to pump water was only of benefit during two or three critical months in most of the above situations. Other farming systems studied had sufficient free time to allow irrigation pumping by humans irrespective of season and pumping demands.

There were only infrequent instances where the use of the automatic pump would prove to be of financial benefit for village water supply. The benefits in providing automatic water supply was only beneficial to women (as they do the majority of the work including water collection) and could be measured in terms of a reduction in their daily workload and freeing them for other activities. In some cases, the women used more energy to perform their daily chores than that consumed per day for the last few months before harvest. The provision of automatic pumping in these cases went towards reducing their energy deficit and could be measured by such factors as decreased morbidity and increased milk yields for lactating mothers.

The only conclusion that can be drawn is that economic analysis falls short when the non financial benefits such as health, wellbeing and reduced drudgery of life are the factors that improve with the provision of a reliable automatic water pump.

8 SUGGESTIONS FOR FUTURE WORK

The work covered to date has encompassed many areas from the conceptual stage and theoretical analysis through to operation of a prototype, concept-proving pump. The viability analysis has shown that for a pump with costs and output as presented in Chapter 7, there is a possibility that with further development this concept may be of practical use in the developing countries for the provision of sanitary village water supplies.

8.1 PHYSICAL IMPROVEMENTS

A characteristic of the step from prototype to final test unit, is that there are several areas which need some development to bring the operating performance and reliability of the pump unit to a satisfactory level. The major areas which need further development are: a valve design for reliable operation under many conditions, development of a composite diaphragm, final body and ancillary design for easy manufacture, maintenance possibilities and the repositioning of some components.

8.1.1 Valve Design

It would be desirable to make the valve both more functional in the diversity of situations where it could be used and to redesign the major sealing surfaces in order to enable the valve to fit into a smaller area. One of the faults of the current valve is its working fluid pressure dependence which may restrict operation to a limited range of working fluid pressures. In order to overcome this problem, the redesign of the valve should incorporate ports from vapour sources and sinks being placed perpendicular to the direction of motion. The spring set-up could remain the same but the means of detention of the shaft could be taken over by detention balls or a similar device. This valve set-up would be a more compact arrangement and could beneficially decrease the dead area in the vapour chamber.

8.1.2 Composite Diaphragm

The use of a composite material for the diaphragm would reduce the dead volume from ballooning effects in both the water and vapour chambers. It would be simple to include reinforcing in a more technical diaphragm manufacturing process than that which was used for the prototype. In larger scale manufacture it would be considered a matter of course for such units.

8.1.3 Body Components Design and Materials

It would be desirable to make the pump body, valve, condenser and collector of materials appropriate for the situations in which they will be used. The process by which the final components are made will have some bearing on the selection of materials for these pieces. The final decision on materials and design will therefore rest in the hands of the developer of the pump into a manufacturable item.

8.1.4 Repositioning of Components

The relocation of the condenser into a separate non-pressurised tank above the top water non-return valve would aid the start up performance of the pump and reduce:

- the corrosion problems faced inside the pump
- difficulties in manufacturing the pump's water jackets
- the number of possible leaks into the water chamber.

The feed pump needs attention as the placement of inlet and outlet ports for the liquid in the feed pump need to be in such an arrangement as to be self priming, i.e. have the liquid inlet at the bottom and the outlet at as high a level as possible.

8.1.5 Maintenance Procedures

Possibly the desire to meet the VLOM ideal for a unit which uses a working fluid other than water is over-ambitious. The VLOM ideal could well fit in with a water pump such as the vapour piston unit. Such units would be less expensive but need a better collector and are only 25% as efficient as the proposed unit. Because the collector is approximately half the cost of the solar pump, the vapour piston unit would be less viable than the proposed set-up.

The final means for the annual maintenance and diaphragm replacement of the pump would have to involve a tiered structure. The structure has been described by some relief agencies as a failure for hand pump schemes. However the problems encountered previously with the tiered structure should not occur if the maintenance is carried out at the specified intervals and the pump is fully reliable till these maintenance dates. Therefore a very reliable pump is necessary.

8.2 A SOLAR PUMP'S FUTURE

With the completion of modifications suggested in the previous section, it would be necessary to conduct several months of testing and endurance trials to ensure reliable operation of the pump. After the unit has been deemed reliable it will be necessary to conduct field trials in situ. The ideal place for such trials would be in one of the South Pacific countries as they have: areas which are relatively poor, large amounts of sunshine and need some improvement in the means of sanitary water supply. Field trials would serve the purpose of getting a user's point of view on how appropriate the technology is, what they may dislike about the system and comparative views on this pump with regard to its alternatives. It is imperative that the village or area in which the units are field-tested show a desire for the units. If not, a wealth of experience in other field trials suggests that the pump will be quickly abandoned at the first sign of extra input or trouble.

The reason for rejection of pump units may not be fully rational by European standards. There have been statements by the villagers (reported by Burgess et al. (1985)) such as 'too foreign' or too much 'imported'. The immediate comments on the solar unit when compared to diesel units was to ignore the savings in fuel costs or increase in productivity. The focus of these observations was on poor technical performance, for example that the solar unit takes a whole day to fill a reservoir where a diesel would only take an hour, or to begrudge the additional manual tasks to be performed.

It is possible that if the villagers are not involved in discussions and appraisals they may consider themselves to be a form of cheap exploitable labour when installing the pump. They may also begrudge the importation of labour to install the unit. The decision making processes about the pump must involve the local people to as large an extent as possible; if not in the pump design, then in the design of storage and drainage areas.

It may be possible for the field testing to be partially sponsored by the Ministry of External Relations and Trade through a research assistance grant or more directly through the New Zealand Bilateral Assistance Programme which has many small project funds (N.Z. Bilateral Assistance Programme, 1990). The project funds are aimed specifically at: enhancing rural living standards, involving women in development, sanitation, water supplies and village facilities.

After field trials, the future of the pump would depend on results and recommendations of the field trials. If positive, the pump's future will depend on the dissemination of information on the pump. The real economics as perceived by the local people who might buy a unit or the value which is placed on the technology by a sponsor or donor of such a water pump, also has a bearing on the future of the solar-thermal water pump.

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APPENDIX 1

Bi-metallic Strip's Efficiency

This program finds the optimum efficiency of a bimetallic strip by changing the dimensions of the strip and the temperature difference undergone in the cycle. The work able to be done by the strip is given by its strain energy.

```
C   BI-METALLIC PROGRAMME
C
    REAL EA,EB,YA,YB,CPA,CPB,STMAX,SBMAX,K1,TA,TB,Y,EI
    REAL W,T0,M,J,K,I,T1,L,EFF,EFF1,THA,THB,LE,TI1
    REAL HEAT,SE,ST,SB,PA,PB,RA
C   SET MATERIALS PROPERTIES
    EB=200E9
    EA=100E9
    YB=4E-6
    YA=24E-6
    CPA=390
    CPB=454
    STMAX=317E6
    SBMAX=690E6
    T0=0
    W=0.01
    EFF1=0
    PA=8000
    PB=6000
C   RUN THROUGH DIMENSION CHOOSING
    DO 733 T1=10,400,10
     WRITE(*,*)T1
     DO 10 I=1,10
       TA=I/1000
      DO 20 J=1,10
        TB=J/1000
       DO 15 K=1,100
         L=K/100
         K1=4+6*TA/TB+4*(TA/TB)**2+EA/EB*(TA/TB)**3
   *+EB/EA*(TB/TA)
         EI=W*TB**3*TA*EB*EA*K1/(12*(TA*EA+TB*EB))
         Y=6*(YB-YA)*(T1-T0)*(TA+TB)/(TB**2*K1)
         RA=(-3)*L**2/(2*L**3)*EI*Y
C    BENDING MOMENT
```

```
          M=RA*L
C   CALCULATION OF THE MAXIMUM STRESSES
          ST=(-6)*M/(W*TB**2*K1)*(2+TB/TA+EA*TA/
 *(EB*TB))-(YB-*YA)*(T1-T0)*EA/K1*(3*TA/TB+2*(TA/TB)**2
 *-EB*TB/(EA*TA))
             SB=6*M/(W*TB**2*K1)*(2+TA/TB+EB*TB/
 *(EA*TA))+(YB-YA)*(T1-T0)*EB/K1*
 *(3*TA/TB+2-EA/EB*(TA/TB)**3)
C   THERMAL INPUT TO CAUSE DEFLECTION
          HEAT=(CPA*W*TA*L*PA+CPB*W*TB*L*PB)*(T1-T0)
C   STRAIN ENERGY OF THE BI-METALLIC STRIP
          SE=RA**2*L**3/(6*EI)
          EFF=SE/HEAT
          IF(ABS(ST).GT.STMAX)THEN
            GOTO20
          ENDIF
          IF(ABS(SB).GT.SBMAX)THEN
            GOTO20
          ENDIF
          IF(ABS(EFF).GT.EFF1)THEN
            THA=TA
            THB=TB
            LE=L
            TI1=T1
            EFF1=ABS(EFF)
            WRITE(*,*)THA,THB,LE,EFF1,TI1
          ENDIF
 15         CONTINUE
 20       CONTINUE
 10     CONTINUE
733 CONTINUE
    WRITE(*,*)THA,THB,LE,EFF1,TI1
```

APPENDIX 2

Rubber's Contraction Efficiency

To calculate the efficiency of a rubber strip producing work from a change in its temperature, the following approach was adopted: A unit of rubber experiences a 350% extension, the work done in extending the rubber is calculated from its strain energy (assuming a constant modulus of elasticity). The change in the rubbers density due to both the 350% extension and the change in temperature on extension (Gough-Joule effect) is ignored as in the complete cycle, the change in density for the contraction part of the cycle will be equal in magnitude but opposite in direction to that of the extension process. The rubber is at a higher temperature than in the initial unstretched position due to both the Gough-Joule effect and hysteresis damping. The rubber then experiences thermal addition of energy, the work done due to the change in density due to temperature change is calculated. The energy extractable from the contraction of the rubber at the new temperature is calculated from the strain energy. The hysteresis damping is taken into account by the addition of the change in temperature of the rubber due to the hysteresis, being added to the change in temperature of the rubber due to the Gough-Joule effect. This combined temperature increase on extension is in the order of 5K. The temperature increase on extension indicates that no work (or negative work) will be done by the system if the difference between the upper and lower temperature limits is less than 5K. All rubber data from Treloar (1958).

For a rubber strip 100mm x 10mm x 10mm extended 350%:

The cold rubber's strain energy is equal to 21J.

For a 100K increase in temperature:

The heated rubber's strain energy is equal to 28.5J.

The work done by the rubber on expansion against the atmosphere due to the temperature increase is given by:

$$W = P \Delta V$$

where:

$$\Delta V = a \times b \times c\,((\,1 + \alpha\,\Delta T\,)^3 - 1\,)$$

W = work, J

P = pressure, Pa

T = temperature, K

V = volume, m^3

a,b,c = the rubber's side dimensions, m

α = thermal expansion, /K

The heat transfer into the rubber is:

$$Q = m\,C_p\,\Delta T$$

where: m = mass of rubber, kg

C_p = specific heat, kJ/kgK

= 21.34 J/K

The work output is:

$$W = 0.75\,\Delta T - 1.013\,((\,1 + \alpha\,\Delta T\,)^3 - 1\,)$$

The efficiency equals the work output divided by heat input:

= W/Q

$\approx$ 0.35% for $5 < \Delta T < 100$

The thermal diffusivity of rubber equals 0.0024 cm^2/sec

APPENDIX 3

Pressure Balance for Double Diaphragm Pump

The use of two diaphragms of different diameters, enclosing two chambers of different volume, allows the change in volume of one chamber to produce a disproportionate change in volume of the second chamber. There is however a pressure balance which must be appropriate to allow this set-up to work. e.g. Fig 41

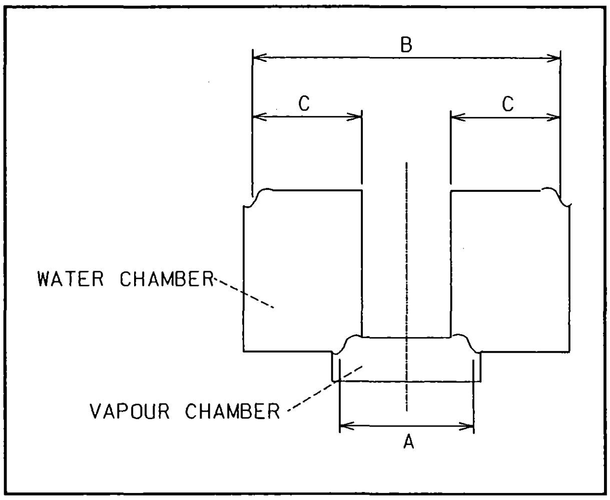

Figure. 41 Pump's Diameter Ratio

The central unit moves within the limits of the diaphragm's travel, for a balance there is atmospheric pressure acting over area B:
say 100 kPa.
There is water pressure acting over area C:
say 50 kPa (water intake, sucking up 5m)
and 130 kPa (water delivery at 3m).
There is vapour pressure acting over area A:
say 185 kPa (collector pressure during water intake)
and 45 kPa (condenser pressure during water delivery).
If the areas of B and A are $0.25m^2$ and $0.1m^2$ then area C is $0.15m^2$
Note: all forces downward are considered positive and all pressures are absolute

so taking in water the pressure balance is:

$100kPa \times .25m^2 - 50kPa \times .15m^2 - 185kPa \times .1m^2 = -1000N$

expelling water the pressure balance is:

$100kPa \times .25m^2 - 130kPa \times .15m^2 - 45kPa \times .1m^2 = 1000N$

The net resulting forces of -1000N and 1000N, acts over area C to give a resultant pressure to accelerate the water into and out of the pump and overcome water frictional and pipe losses.

The collector temperature will relate to the collector pressure for the working fluid used. The condenser temperature will relate to the condenser pressure for the working fluid used.

It is most probable that one will work from the fluid pressure at the condenser and collector temperatures and then calculate the diameter ratios of A to B to pump water to and from the desired heights.

APPENDIX 4

Control Mass Exergy Discussion

The change in exergy equation for a control mass can be written (Moran, 1982)

$$\Delta\phi = \int_1^2 (1 - \frac{T_o}{T_s}) \, \delta Q - (W - P_o \, \delta V) - I \qquad \text{A4.1}$$

(i) (ii) (iii)

where:
T = temperature, K
Q = heat transfer, kJ
W = work transfer, kJ
P = pressure, kPa
V = volume, m^3
I = irreversibilities, kJ
$_o$ indicates base conditions
$_s$ indicates current position

where (i) represents the availability flow with heat transfer, (ii) is availability flow with work interactions and (iii) is the destruction of exergy due to irreversibilities. For work performed in a constant pressure increase in volume of the control mass, W equates to P ΔV . For this situation there is work done by the system on the boundary by a pressure equivalent to P, where P > P_o. ΔV is positive so the term Po ΔV is positive, indicating that work has been done by the system (on the environment at Po). In terms of exergy the amount of work able to be performed by the system is reduced by (W - Po ΔV), this indicating that for an amount of work P ΔV, Po ΔV is available to do useful work because it is effectively in storage in the atmosphere or environment, and is able to do work of Po ΔV in contracting the system should the system pressure P drop below Po.

APPENDIX 5

Computer Fortran Files

```
C
C
C    OPTIMISATION BASE PROGRAMME
C    FOR WATER PUMP WITH ONE WORKING FLUID
C    GOING THROUGH A DAILY CYCLE WITH INTERMITENT CLOUD
COVER
C
C
C
C
      IMPLICIT REAL(A-T)
      IMPLICIT INTEGER(W-Z)
      INTEGER MONNUM,DAYNUM,I,N,LSIGN
      REAL LATDEG,ORIENT,U,V,W,SOLRAD,COLALT,P(80),WATOUT
      REAL HERP,SRADSK,SRADGR,FSS,FSG,TRANS,SORB,THICK,MAXRAD
      REAL
TLOW,DULAST,TOGLAST,PLTEMP1,PLTEMP2,PLTEMP3,CPL,CPM,CPG
      REAL MM,ML,MG,HFG,TET,CC(80),DIFF(80),BET(80),DIF
      REAL
HR(80),TORADD(80),TORAD(80),ABCOEF,HOUR,WPMM,NUMB,COLAZI
      REAL MONTH,BETA,A,B,C,D,E,F,UTL,UBL,MAXT,THERCO,REPEAT
      REAL
DFLOW,ONE,ZER,TOR,OMTOR,GOLD,VALTOR,VALOMTOR,RI,VIL,CU,CV,
CW
      REAL RA,RL,DIA,ALTIT(80),DU,TOG,TOT,CU2,CV2,CW2
      REAL DIAM,RATI,RLEN,LIN,SUKH,PREH,CONDV,CONDA,SPRF,ATEMP
      REAL SOAD(80),HEP(80),AS(80),BB(80),EE(80),UIN,VALOUT(80)
      REAL DAYDIF,SEADIF,TAMB,EMITP,EMITG,OUTCON,TA
      CHARACTER STRING(38)*120
*$include GKSDEFN.FOR
C
C   INPUT THE STRING OF QUESTIONS
C
      STRING(1) =  'WHICH HEMISPHERE (NORTH OR SOUTH)? 1=NORTH'
      STRING(2) =  'LATITUDE IN DEGREES (0 TO 90)?'
      STRING(3) =  'WHAT IS THE MONTH (1 TO 12)?'
      STRING(4) =  'WHAT IS THE DAY OF THE MONTH?'
      STRING(5) =  'LOCAL SOLAR TIME (24 HOUR/DECIMAL FORMAT)?'
      STRING(6) =  'WHAT IS THE COLLECTORS ALTITUDE ANGLE?'
      STRING(7) =  'WHAT IS THE COLLECTORS AZIMUTH ANGLE?'
      STRING(8)=  'HOW MANY GLASS PANELS DOES THE COLLECTOR
HAVE?'
      STRING(9)=  'WHAT IS THE REFRACTIVE INDEX FOR THE GLASS?'
      STRING(10)=  'WHAT IS THE ABSORPTION COEFFICIENT OF THE
GLASS?'
      STRING(11)=  'WHAT IS THE GLASS THICKNESS? (M)'
      STRING(12)=  'WHAT IS THIS COMPUTATION RUN NUMBER?'
```

```
      STRING(13)=  'WHAT IS THE MAXIMUM AVERAGE MAXIMUM
TEMPERATURE IN
     *THE HOTTEST SEASON? (C)'
      STRING(14)=  'WHAT IS THE RANGE OF DAILY TEMPERATURES?
ie
     *AVERAGE DAILY MAXIMUM MINUS AVERAGE DAILY MINIMUM'
      STRING(15)=  'WHAT IS THE RANGE OF SEASONAL TEMPERATURE
VARIATION?
     * ie HOTEST AVERAGE MAX MINUS COOLEST AVERAGE MAX OVER
THE YEAR'
      STRING(16)=  'WHAT IS THE SPACING BETWEEN THE GLASS
PLATES? (M)'
      STRING(17)=  'WHAT IS THE GLASS PLATES EMITTANCE?'
      STRING(18)=  'WHAT IS THE PLATES EMITTANCE?'
      STRING(19)=  'WHAT IS THE PLATE TEMPERATURE? (C)'
      STRING(20)=  'WHAT IS THE OUTER CONDUCTIVITY?'
      STRING(21)=  'CONDUCTIVITY OF THE BACK INSULATION?'
      STRING(22)=  'THICKNESS OF BACK INSULATION? (M)'
      STRING(23) =  'assumed atmospheric temperature'
      STRING(24) =  'Initial pump diameter'
      STRING(25) =  'diameter ratio'
      STRING(26) =  'initial rubber length'
      STRING(27) =  'lower input power'
      STRING(28) =  'upper input power'
      STRING(29) =  'suction head'
      STRING(30) =  'pressure head'
      STRING(31) =  'condenser volume'
      STRING(32) =  'condenseer area'
      STRING(33) =  'spring force'
      STRING(34) =  'DID YOU MAKE AN ERROR 1=YES PLEASE
RE-ENTER ALL V
     *ALUES'
      STRING(35) =  'WOULD YOU LIKE TO CHANGE FROM THE PRESET
VALUES?
     *1.0=YES  2.0=NO'
      STRING(36) =  'WOULD YOU LIKE TO USE THE PROGRAMME AGAIN
1=YES'
      STRING(37) =  'CLOUDINESS (0-1)'
      STRING(38) =  'TOTAL SOLAR RADIATION'
C
C   ENTER SOME DEFAULT RESPONSES
C
    9 REPEAT=1
      LSIGN=-1
      LATDEG=43
      W=3
      X=1
```

```
      HOUR=4
      COLALT=56
      COLAZI=2
      N=2
      REFG=1.586
      ABCOEF=.04
      THICK=.001
      NUMB=123
      MAXT=23
      DAYDIF=10
      SEADIF=10
      SPACE=.004
      EMITG=.84
      EMITP=.80
      PTEMP=68
      OUTCON=25
      THERCO=.06
      THICKB=.03
      DIAM=.24
      RATI=1.39
      RLEN=.04
      LIN=300
      UIN=800
      SUKH=6.5
      PREH=1.5
      CONDV=.002
      CONDA=.4
      SPRF=300
      ATEMP=17
      VALTO=0
      VALOMTO=0
      MAXRAD=0
      TSR=1.68E7
      CLO=0.5
C
C   CALL THE CLEARSCREEN AND QUERY SUBROUTINES
C   BUT ONLY IF THE VALUES ARE TO BE CHANGED FROM THE
DEFAULT VALUES
C
   11 CALL SETSCR(2)
      WRITE(*,*) STRING(35)
      READ(*,*) REPEAT
      IF(REPEAT.EQ.2) THEN
        GOTO320
      ENDIF
      IF(REPEAT.EQ.1) THEN
        GOTO40
```

```
      ENDIF
C
C  HEMISPHERE CHECK
C
   40 WRITE(*,*) STRING(1),LSIGN
      READ(*,*) LSIGN
      IF(LSIGN.NE.1) THEN
         LSIGN=-1
      ENDIF
C  CHECK THE LATITUDE
      WRITE(*,*) STRING(2),LATDEG
      READ(*,*) LATDEG
C
C   CHECK VALUES FOR THE MONTH
C
      WRITE(*,*) STRING(3),W
      READ(*,*) W
      MONTH=W
      MONNUM=W
C  CHECK THE DAY OF MONTH
C
      WRITE(*,*) STRING(4),X
      READ(*,*) X
      DAY = X
      DAYNUM = X
C
C  CHECK SOLAR TIME
C
      WRITE(*,*) STRING(5),HOUR
      READ(*,*) HOUR
C
C  CHECK THE COLLECTORS ALT + AZI ANGLES
C
      WRITE(*,*) STRING(6),COLALT
      READ(*,*) COLALT
C
      WRITE(*,*) STRING(7),COLAZI
      READ(*,*) COLAZI
C
C  GET THE NUMBER OF GLASS PANES
C
      WRITE(*,*) STRING(8),N
      READ(*,*) N
C
C  GET THE REFRACTIVE INDEX FOR GLASS
C
      WRITE(*,*) STRING(9),REFG
```

```
      READ(*,*) REFG
C
C   GET THE ABSORPTION COEFF FOR THE GLASS
C
      WRITE(*,*) STRING(10),ABCOEF
      READ(*,*) ABCOEF
C
C   GET THE GLASS THICKNESS
C
      WRITE(*,*) STRING(11),THICK
      READ(*,*) THICK
C
C   GET THE RUN NUMBER
C
      WRITE(*,*) STRING(12),NUMB
      READ(*,*) NUMB
C
C   MAXIMUM TEMP
C
      WRITE(*,*) STRING(13),MAXT
      READ(*,*) MAXT
C
C   DAILY TEMP DIFF
C
      WRITE(*,*) STRING(14),DAYDIF
      READ(*,*) DAYDIF
C
C   SEASONAL TEMP DIFF
C
      WRITE(*,*) STRING(15),SEADIF
      READ(*,*) SEADIF
C
C   DISTANCE BETWEEN PLATES
C
      WRITE(*,*) STRING(16),SPACE
      READ(*,*) SPACE
C
C   GLASS EMITANCE
C
      WRITE(*,*) STRING(17),EMITG
      READ(*,*) EMITG
C
C   PLATE EMITTANCE
C
      WRITE(*,*) STRING(18),EMITP
      READ(*,*) EMITP
C
```

```
C  PLATE TEMPERATURE
C
      WRITE(*,*) STRING(19),PTEMP
      READ(*,*) PTEMP
      PTEMP=PTEMP+273
C
C  OUTWARD CONDUCTIVITY OF GLASS - AIR INTERFACE
C
      WRITE(*,*) STRING(20),OUTCON
      READ(*,*) OUTCON
C
C  CODUCTIVITY OF BOTTOM INSULATION
C
      WRITE(*,*) STRING(21),THERCO
      READ(*,*) THERCO
C
C  THICKNESS OF BACK INSULATION
C
      WRITE(*,*) STRING(22),THICKB
      READ(*,*) THICKB
C
C  DIAMETER
C
      WRITE(*,*) STRING(24),DIAM
      READ(*,*) DIAM
C
C  RATIO
C
  250 WRITE(*,*) STRING(25),RATI
      READ(*,*) RATI
      IF(RATI.LE.0.OR.RATI.GT.3) THEN
        GOTO250
      ENDIF
C
C  RUBBER LENGTH
C
  260 WRITE(*,*) STRING(26),RLEN
      READ(*,*) RLEN
      IF(RLEN.LT.0.OR.RLEN.GT.1) THEN
        GOTO260
      ENDIF
C
C  LOWER ENERGY IN
C
      WRITE(*,*) STRING(27),LIN
      READ(*,*) LIN
C
```

```
C   UPPER ENERGY IN
C
      WRITE(*,*) STRING(28),UIN
      READ(*,*) UIN
C
C   SUCTION HEAD
C
      WRITE(*,*) STRING(29),SUKH
      READ(*,*) SUKH
C
C   PRESSURE HEAD
C
      WRITE(*,*) STRING(30),PREH
      READ(*,*) PREH
C
C   CONDENSER VOLUME
C
      WRITE(*,*) STRING(31),CONDV
      READ(*,*) CONDV
C
C   CONDENSER AREA
C
      WRITE(*,*) STRING(32),CONDA
      READ(*,*) CONDA
C
C   SPRING FORCE
C
      WRITE(*,*) STRING(33),SPRF
      READ(*,*) SPRF
C
C   ASSUMED ATMOSPHERIC TEMPERATURE
C
      WRITE(*,*) STRING(23),ATEMP
      READ(*,*) ATEMP
C
C   CLOUD COVER
C
      WRITE(*,*) STRING(37),CLO
      READ(*,*) CLO
C
C   TOTAL SOLAR RADIATION
C
      WRITE(*,*) STRING(38),TSR
      READ(*,*) TSR
C
C   ERROR RECOVERY
C
```

```
      WRITE(*,*) STRING(34)
      READ(*,*) ERR
      IF(ERR.EQ.1)THEN
        GOTO11
      ENDIF
C
 320 CLOSE(1)
      IF(REPEAT.EQ.2) THEN
        PTEMP=PTEMP+273
        DAY = X
        MONTH=W
        MONNUM=W
        DAYNUM = X
      ENDIF
C
C   FIND THE ALTITUDE AND AZIMUTH ANGLES WITH A SUBROUTINE
C
C
      LATDEG=LATDEG*LSIGN
      LATRAD=LATDEG*PI/180.0
C
C   WORKING OUT VARIABLES FOR THE WHOLE DAY, IE. 80 * .25 HOUR
STEPS
C
 340 DO 46 I=1,80
C
       CALL
SUNAG(ORIENT,LATRAD,MONNUM,DAYNUM,HOUR,BETA,GAMMA,U,V,W
)
C
        ALTIT(I)=W*(-1)
        IF(ALTIT(I).LT.0)THEN
          ALTIT(I)=0
        ENDIF
        IF(W.GE.0) THEN
          TORAD(I)=0
          TORADD(I)=0
          SOAD(I)=0
          DIFF(I)=0
          HR(I)=HOUR
          HOUR=HOUR+.25
          GOTO46
        ENDIF
C
C   INTERPOLATING THE VALUES FOR THE CONSTANTS TO FIND
SOLAR RADIATION INTENSITY
C
```

```
      IF (ABS(DAY-21).EQ.(DAY-21)) THEN
        GOTO44
      ENDIF
      CALL COEFF(MONTH,A,B,E)
      F=E
      C=A
      D=B
      MONTH=MONTH-1
      CALL COEFF(MONTH,A,B,E)
      A=C+((21-DAY)/30)*(A-C)
      B=D+((21-DAY)/30)*(B-D)
      E=F+((21-DAY)/30)*(E-F)
      MONTH=MONTH+1
      GOTO42
 44    CALL COEFF(MONTH,A,B,E)
      C=A
      D=B
      F=E
      MONTH=MONTH+1
      CALL COEFF(MONTH,A,B,E)
      A=A+((30-(DAY-21))/30)*(C-A)
      B=B+((30-(DAY-21))/30)*(D-B)
      E=E+((30-(DAY-21))/30)*(F-E)
      MONTH=MONTH-1
C
C  SOLAR RADIATION INTENSITY AT A NORMAL TO THE DIRECTION
OF THE RAYS
C
 42   SOLRAD=A/(EXP(B/SIN(BETA)))
C
C  DIFFUSE SOLAR RADIATION FROM THE SKY
C
      WPMM=0
      GLASSORB=0
      COLALTR=COLALT*PI/180
      COLAZIR=COLAZI*PI/180
C
      FSS=(1+COS((PI/2)-COLALTR))/2
      SRADSK=E*SOLRAD*SIN(BETA)*FSS
C  ACCOUNT FOR TRANSMITANCE AND ABSORBANCE IN GLASS
      HERP=59.68-.1388*(COLALT)+.001497*(COLALT)**2
      HERPR=HERP*PI/180
      CALL GLASS(N,TRANS,TRINT4,SORB,HERPR,THICK,ABCOEF,REFG)
      TRANS1=TRINT4*SRADSK
      SORB1=SORB*SRADSK
      WPMM=WPMM+TRANS1
      GLASSORB=GLASSORB+SORB1
```

```
C
C   DIFFUSE SOLAR RADIATION REFLECTED FROM THE GROUND
C
      FSG=1-FSS
      SRADGR=.2*FSG*(SOLRAD*SIN(BETA)*(E+SIN(BETA)))
C   ACCOUNT FOR TRANSMITANCE AND ABSORBANCE IN GLASS
      HERP=90-.5788*(COLALT)+.002693*(COLALT)**2
      HERPR=HERP*PI/180
      CALL GLASS(N,TRANS,TRINT5,SORB,HERPR,THICK,ABCOEF,REFG)
      TRANS2=TRINT5*SRADGR
      SORB2=SORB*SRADGR
      WPMM=WPMM+TRANS2
      GLASSORB=GLASSORB+SORB2
C
C   COLLECTOR DIRECTION CALCULATIONS
C
      CU=COS(COLAZIR)*COS(COLALTR)
      CV=SIN(COLAZIR)*COS(COLALTR)
      CW=SIN(COLALTR)
C
C   ANGLE BETWEEN NORMAL TO COLLECTOR AND SUNS RAYS
C
      CU2=CU**2
      CV2=CV**2
      CW2=CW**2
      HERPR=ACOS((CU*U+CV*V+CW*W)/(SQRT(U*U+V*V+W*W)*
    *SQRT(CU2+CV2+CW2)))
      CU=SQRT(CU)
      CV=SQRT(CV)
      CW=SQRT(CW)
      HERPR=PI-HERPR
C
C   CALL A PROGRAMME TO FIND THE TOTAL DIRECT SOLAR
RADIATION ON THE SURFACE
C     BEHIND THE GLAZING
C   ACCOUNTING FOR TRANSMITANCE AND ABSORBANCE IN GLASS
C
      CALL GLASS(N,TRANS6,TRINT,SORB,HERPR,THICK,ABCOEF,REFG)
      TRANS3=TRANS6*SOLRAD*COS(HERPR)
      SORB3=SORB*SOLRAD*COS(HERPR)
      WPMM=WPMM+TRANS3
      GLASSORB=GLASSORB+SORB3
C
C
C   FIND TA (AMBIENT TEMP) FOR ANY PLACE AT ANY TIME OF ANY
DAY
C
```

```
      CALL
AMBIENT(HOUR,DAYNUM,MONNUM,LSIGN,DAYDIF,SEADIF,MAXT,
   *TAMB,LATDEG)
      TA=TAMB+273
C
C  HEAT LOSS THROUGH GLAZING (TOP LOSS)
C  FIND ULT (TOP LOSS COEFF) INCLUDING RADIATIVE LOSSES
C
      CALL
TOLOSS(TA,SPACE,EMITP,N,OUTCON,EMITG,PTEMP,COLALTR,
   *UTL)
C
C  ACCOUNT FOR BOTTOM LOSS (SIDE LOSSES IGNORED)
C
      UBL=THICKB*THERCO
      UTL=UTL+UBL
C
C  TOTAL HEAT LOSS
C
      QLOSS=UTL*(PTEMP-TA)
C
C   CLOUDINESS CALCULATIONS
C   CORRECTION FOR CLOUDINESS OBSERVATION
C
    CG=CLO
    CS=.965*CG-.12
C
C   CALCULATE DIFFUSE RADIATION LEVEL DURING CLOUDY PERIOD
C
    FICTOR=ABS(SIN((HOUR-6)*PI/12))
    CC(I)=(CG*FICTOR+CS*(1-FICTOR))+.24-.0018*LATDEG
    CD=CC(I)
    CALL CLCO(CD,DIF)
    DIFF(I)=DIF*SOLRAD
C
C  CHECH NOS
C
      SOAD(I)=SOLRAD
      HEP(I)=QLOSS
      AS(I)=TAMB
      BB(I)=UTL-UBL
      EE(I)=UBL
      BET(I)=BETA
C
      HR(I)=HOUR
      TORADD(I)=WPMM
      TORAD(I)=SOAD(I)*COS(HERPR)
```

```
        HOUR=HOUR+.25
        IF(TORAD(I).GT.MAXRAD)THEN
          MAXRAD=TORAD(I)
        ENDIF
  46 CONTINUE
      IF(MAXRAD.GT.UIN)THEN
        UIN=MAXRAD
      ENDIF
C
C   CLOUDINESS CALCULATIONS TO FIND PERIOD OF CLOUDINESS PER
.25 HOUR
C
      DU=0
      TOT=0
      TOG=TSR
      DO 1357 T=0,15
        TOGLAST=TOT
        TOT=0
        DO 1337 I=1,80
          TOT=TOT+(SOAD(I)*ALTIT(I)*(15-T)*60+DIFF(I)*T*60)
          IF(I.EQ.80.AND.TOT.LT.TSR)THEN
            GOTO1358
          ENDIF
 1337   CONTINUE
 1357 CONTINUE
 1358 T=T-(1-((TOGLAST-TSR)/(TOGLAST-TOT)))
C
C  RUN THROUGH OPTIMISATION ROUTINE FOR 5
DIFFERENTRADIATION CONDITIONS
C  AND PUT RESULTS ON DISC
C
      TOR=0
      RI=0
      DIA=0
      RA=0
      RL=0
      GOLD=(UIN-100-(LIN+100))/4
      UIN=UIN-100
      CALL
COPD(UIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *AS,TORAD,VALOUT,VALTO,P,UIN,VALTOR,DIF,UTL,T)
      OPEN(4,FILE='B:\RESULT1.DAT',STATUS='UNKNOWN')
      WRITE(4,50) DIAM,RATI,RLEN,SPRF,CONDA,CONDV
      WRITE(4,55) PREH,SUKH,ATEMP,VALTOR,UIN
      CLOSE(4,STATUS='KEEP')
      WRITE(*,*)UIN
      IF(VALTOR.GT.TOR)THEN
```

```
          TOR=VALTOR
          RI=UIN
          DIA=DIAM
          RA=RATI
          RL=RLEN
        ENDIF
        UIN=UIN-GOLD
        CALL
COPD(UIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *AS,TORAD,VALOUT,VALTO,P,UIN,VALTOR,DIF,UTL,T)
        OPEN(4,FILE='B:\RESULT2.DAT',STATUS='UNKNOWN')
        WRITE(4,50) DIAM,RATI,RLEN,SPRF,CONDA,CONDV
        WRITE(4,55) PREH,SUKH,ATEMP,VALTOR,UIN
        CLOSE(4,STATUS='KEEP')
        WRITE(*,*)UIN
        IF(VALTOR.GT.(.99*TOR))THEN
          TOR=VALTOR
          RI=UIN
          DIA=DIAM
          RA=RATI
          RL=RLEN
        ENDIF
        UIN=UIN-GOLD
        CALL
COPD(UIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *AS,TORAD,VALOUT,VALTO,P,UIN,VALTOR,DIF,UTL,T)
        OPEN(4,FILE='B:\RESULT3.DAT',STATUS='UNKNOWN')
        WRITE(4,50) DIAM,RATI,RLEN,SPRF,CONDA,CONDV
        WRITE(4,55) PREH,SUKH,ATEMP,VALTOR,UIN
        CLOSE(4,STATUS='KEEP')
        WRITE(*,*)UIN
        IF(VALTOR.GT.(.99*TOR))THEN
          TOR=VALTOR
          RI=UIN
          DIA=DIAM
          RA=RATI
          RL=RLEN
        ENDIF
        UIN=UIN-GOLD
        CALL
COPD(UIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *AS,TORAD,VALOUT,VALTO,P,UIN,VALTOR,DIF,UTL,T)
        OPEN(4,FILE='B:\RESULT4.DAT',STATUS='UNKNOWN')
        WRITE(4,50) DIAM,RATI,RLEN,SPRF,CONDA,CONDV
        WRITE(4,55) PREH,SUKH,ATEMP,VALTOR,UIN
        CLOSE(4,STATUS='KEEP')
        WRITE(*,*)UIN
```

```
      IF(VALTOR.GT.(.99*TOR))THEN
        TOR=VALTOR
        RI=UIN
        DIA=DIAM
        RA=RATI
        RL=RLEN
      ENDIF
      UIN=LIN+100
      CALL
COPD(UIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *AS,TORAD,VALOUT,VALTO,P,UIN,VALTOR,DIF,UTL,T)
      OPEN(4,FILE='B:\RESULT5.DAT',STATUS='UNKNOWN')
      WRITE(4,50) DIAM,RATI,RLEN,SPRF,CONDA,CONDV
      WRITE(4,55) PREH,SUKH,ATEMP,VALTOR,UIN
      CLOSE(4,STATUS='KEEP')
      WRITE(*,*)UIN
      IF(VALTOR.GT.(.99*TOR))THEN
        TOR=VALTOR
        RI=UIN
        DIA=DIAM
        RA=RATI
        RL=RLEN
      ENDIF
C
C  RUN THROUGH OPTIMUM TO GIVE VALUES OF PERFORMANCE FOR
EACH .25 HOUR
C
      CALL COD(RI,PREH,SUKH,DIA,RA,RL,ATEMP,SPRF,
     *CONDA,CONDV,AS,TORAD,VALOUT,WATOUT,P,DIF,UTL,T)
        DFLOW=0
        DO 1820 I=1,80
          IF(P(I).EQ.0)THEN
            GOTO1822
          ENDIF
          VALOUT(I)=VALOUT(I)*18000/P(I)
 1822     DFLOW=VALOUT(I)+DFLOW
 1820 CONTINUE
C
C  OUTPUT THE OPTIMUM PUMPS RESULTS TO FILE ON DISC A
C
 2010 OPEN(4,FILE='B:\RESULTS.DAT',STATUS='UNKNOWN')
      WRITE(4,43) NUMB,LATDEG,MONTH
      WRITE(4,45) DAY,COLAZI,COLALT
      WRITE(4,52)
      WRITE(4,50) DIA,RA,RL,SPRF,CONDA,CONDV
      WRITE(4,53)
      WRITE(4,51) PREH,SUKH,ATEMP,VALTOR,DFLOW,RI
```

```
      WRITE(4,47)
      DO 49 I = 1,80
        P(I)=P(I)/20
        WRITE(4,48) HR(I),TORAD(I),SOAD(I),HEP(I),AS(I),VALOUT(I),P(I)
   49 CONTINUE
   43 FORMAT(1X,'RUN NO =',F5.0,'   LAT = ',F6.2,'   MONTH = ',F5.0)
   45 FORMAT(1X,'DAY = ',F5.0,'   COLAZI = ',F6.2,'   COL ALT = ',F5.2)
   47 FORMAT(2X,'HOUR  RADIATION  SOLRAD  QLOSS  TAMB  m3/MIN
TIME')
   48 FORMAT(2X,F5.2,2X,F6.1,2X,F6.1,2X,F6.1,2X,F5.1,2X,F5.1,2X,F7.2)
   52
FORMAT(1X,'DIAM=',2X,'RATIO=',2X,'RLEN=',2X,'SPRF=',2X,'CONDA='
     *,2X,'CONDV=')
   50 FORMAT(1X,F6.4,2X,F4.2,2X,F4.2,2X,F7.1,2X,F5.2,2X,F5.2)
   51 FORMAT(1X,F4.1,2X,F4.1,2X,F5.1,2X,F5.1,2X,F5.2,2X,F6.1)
   55 FORMAT(1X,F4.1,2X,F4.1,2X,F5.1,2X,F5.1,2X,F6.1)
   53 FORMAT(1X,'PREH=',2X,'SUKH=',2X,'ATEMP=',2X,'WATOUT=   1
DAYS
     *WATER=     DESIGN POWER')
       CLOSE(4,STATUS='KEEP')
C
C   CHECK TO SEE IF USER WANTS TO USE PROGRAM AGAIN
C
      WRITE(*,*) STRING(36)
      WRITE(*,*)'OPTIMISATION IS FINISHED YOU MAY TURN THE
MACHINE OFF'
      READ(*,*) AGAIN
      IF(AGAIN.EQ.1)THEN
        GOTO 9
      ENDIF
C   SET THE SCREEN TO 80x25 MONOCHROME MODE
      CALL SETSCR(2)
      STOP
      END

      SUBROUTINE CLCO(CC,DIFF)
      REAL CC,DIFF
C
C   LOOK UP TABLE FOR DIFFUSE FACTOR VERSUS CLOUD COVER
C
      IF(CC.GE.0.AND.CC.LT..5)THEN
        DIFF=.2
      ENDIF
      IF(CC.GE..5.AND.CC.LT.1.5)THEN
        DIFF=.26
      ENDIF
      IF(CC.GE.1.5.AND.CC.LT.2.5)THEN
```

```
        DIFF=.32
      ENDIF
      IF(CC.GE.2.5.AND.CC.LT.3.5)THEN
        DIFF=.35
      ENDIF
      IF(CC.GE.3.5.AND.CC.LT.4.5)THEN
        DIFF=.38
      ENDIF
      IF(CC.GE.4.5.AND.CC.LT.5.5)THEN
        DIFF=.41
      ENDIF
      IF(CC.GE.5.5.AND.CC.LT.6.5)THEN
        DIFF=.45
      ENDIF
      IF(CC.GE.6.5.AND.CC.LT.7.5)THEN
        DIFF=.46
      ENDIF
      IF(CC.GE.7.5.AND.CC.LT.8.5)THEN
        DIFF=.47
      ENDIF
      IF(CC.GE.8.5.AND.CC.LT.9.5)THEN
        DIFF=.45
      ENDIF
      IF(CC.GE.9.5.AND.CC.LE.10)THEN
        DIFF=.43
      ENDIF
      RETURN
      END

      SUBROUTINE
COD(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,
     *CONDV,AS,TORAD,ILUT,WATOUT,IP,DIF,UTL,TIME)
      REAL DIAM,RATI,RLEN,SUKH,PREH,CONDV,CONDA,SPRF,ATEMP
      REAL IP(80),ILUT(80),AS(80),TORAD(80),WATOUT,WIN
      REAL LUT,TORAE,DIF,UTL,TIME
      INTEGER B,I
C
C  SUBROUTINE TO WORK THROUGH ONE DAYS WATER PUMPING IN
.25 HOUR INTERVALS
C
      WATOUT=0
      DO 3050 I=1,80
        IF(TORAD(I).LE.200)THEN
          ILUT(I)=0
          IP(I)=0
          GOTO3050
        ENDIF
```

```
      WRITE(*,*)I
      TORAE=TORAD(I)
      AB=AS(I)
      CALL CPOG(TORAE,PREH,SUKH,DIAM,RATI,RLEN,AB,SPRF,CONDA,
   *  CONDV,LUT,P,DIF,UTL,TIME)
      WATOUT=LUT
      ILUT(I)=LUT
      IP(I)=P
 3050 CONTINUE
    RETURN
    END

    SUBROUTINE
CODG(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,
   *CONDV,AS,TORAD,ILUT,WATOUT,IP,DIF,UTL,TIME)
    REAL DIAM,RATI,RLEN,SUKH,PREH,CONDV,CONDA,SPRF,ATEMP
    REAL IP(80),ILUT(80),AS(80),TORAD(80),WATOUT,WIN
    REAL LUT,TORAE,DIF,UTL,TIME
    INTEGER B,I
C
C  SUBROUTINE TO WORK THROUGH ONE DAYS WATER PUMPING IN
.5 HOUR INTERVALS
C
    WATOUT=0
    DO 3050 I=1,80,2
      IF(TORAD(I).LE.200)THEN
        ILUT(I)=0
        IP(I)=0
        GOTO3050
      ENDIF
      WRITE(*,*)I
      TORAE=TORAD(I)
      AB=AS(I)
      CALL CPOG(TORAE,PREH,SUKH,DIAM,RATI,RLEN,AB,SPRF,CONDA,
   *  CONDV,LUT,P,DIF,UTL,TIME)
      WATOUT=LUT
      ILUT(I)=LUT
      IP(I)=P
 3050 CONTINUE
    RETURN
    END

    SUBROUTINE
COPD(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,
   *CONDV,AS,TORAD,ILUT,WATOUT,IP,RI,DFLOW,DIF,UTL,TIME)
    REAL DIAM,RATI,RLEN,SUKH,PREH,CONDV,CONDA,SPRF,ATEMP
    REAL DFLOW,VALOUT(80),OP(80),P,DOR,DIF,UTL,TIME
```

```
      REAL FACT,IP(80),ILUT(80),AS(80),TORAD(80),MAXOUT,WATOUT,WIN
      REAL LUT,RB,F1,F2,F3,F4,F5,F6,TORAE,RI,G
      INTEGER B,I
C
C  SEARCH PROGRAMME FOR OPTIMUM VARIABLE VALUES
C
      VOL=0
      GOG=0
      G=0
      DOR=1
      MAXOUT=0
      WATOUT=0
C
C  VARIABLE CHANGE FACTOR CALCULATIONS
C
      FACT=0.25
 3010 FACT=FACT-.05
      IF(DOR.EQ.2)THEN
        GOTO4000
      ENDIF
      IF(FACT.LE.0.049.AND.DOR.EQ.1)THEN
        DOR=DOR+1
        FACT=.025
      ENDIF
C
C  RATIO VARIATION UP AND DOWN BY CHANGE FACTOR
C
 3020 gog=0
      F1=FACT*RATI
      B=0
      RATI=RATI+F1
      IF(RATI.GT.5)THEN
        GOTO3222
      ENDIF
      CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *LUT,P,TORAD,AS,DIF,UTL,TIME)
      IF(LUT.GT.MAXOUT)THEN
        B=1
        MAXOUT=LUT
        RB=B*1000
        WRITE(*,*)RB
      ENDIF
 3222 RATI=RATI-2*F1
      IF(RATI.LE.1)THEN
        GOTO3330
      ENDIF
```

```
      CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *LUT,P,TORAD,AS,DIF,UTL,TIME)
      IF(LUT.GT.MAXOUT)THEN
        B=2
        MAXOUT=LUT
        RB=B*1000
        WRITE(*,*)RB
      ENDIF
 3330 RATI=RATI+F1
      IF(B.EQ.1.OR.B.EQ.2)THEN
        F2=FACT*DIAM
        DIAM=DIAM+F2
        IF(DIAM.GT..5)THEN
          GOTO3201
        ENDIF
      CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *LUT,P,TORAD,AS,DIF,UTL,TIME)
        IF(LUT.GT.MAXOUT)THEN
          GOG=15
          MAXOUT=LUT
          RB=B*1000
          WRITE(*,*)RB
        ENDIF
 3201   DIAM=DIAM-2*F2
        IF(DIAM.LT..1)THEN
          GOTO3211
        ENDIF
      CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *LUT,P,TORAD,AS,DIF,UTL,TIME)
        IF(LUT.GT.MAXOUT)THEN
          GOG=16
          MAXOUT=LUT
          RB=B*1000
          WRITE(*,*)RB
        ENDIF
 3211   DIAM=DIAM+F2
      ENDIF
      F2=FACT*DIAM
      DIAM=DIAM+F2
      IF(DIAM.GT..5)THEN
        GOTO3200
      ENDIF
      CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
```

```
     *LUT,P,TORAD,AS,DIF,UTL,TIME)
     IF(LUT.GT.MAXOUT)THEN
       B=3
       MAXOUT=LUT
       RB=B*1000
       WRITE(*,*)RB
     ENDIF
 3200 DIAM=DIAM-2*F2
     IF(DIAM.LT..1)THEN
       GOTO3210
     ENDIF
     CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *LUT,P,TORAD,AS,DIF,UTL,TIME)
     IF(LUT.GT.MAXOUT)THEN
       B=4
       MAXOUT=LUT
       RB=B*1000
       WRITE(*,*)RB
     ENDIF
 3210 DIAM=DIAM+F2

     F3=FACT*RLEN
     RLEN=RLEN+F3
     IF(RLEN.GT..05)THEN
       GOTO3220
     ENDIF
     CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *LUT,P,TORAD,AS,DIF,UTL,TIME)
     IF(LUT.GT.MAXOUT)THEN
       B=5
       MAXOUT=LUT
       RB=B*1000
       WRITE(*,*)RB
     ENDIF
 3220 RLEN=RLEN-2*F3
     IF(RLEN.LT..01)THEN
       GOTO3230
     ENDIF
     CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
     *LUT,P,TORAD,AS,DIF,UTL,TIME)
     IF(LUT.GT.MAXOUT)THEN
       B=6
       MAXOUT=LUT
       RB=B*1000
```

```
       WRITE(*,*)RB
     ENDIF
 3230 RLEN=RLEN+F3
C     F5=FACT*CONDV
C     CONDV=CONDV+F5
C     IF(CONDV.GT..5)THEN
C        GOTO3110
C     ENDIF
C     CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
C    *LUT,P,TORAD,AS,DIF,UTL,TIME)
C     IF(LUT.GT.MAXOUT)THEN
C        B=9
C        MAXOUT=LUT
C        RB=B*1000
C        WRITE(*,*)RB
C     ENDIF
C 3110 CONDV=CONDV-2*F5
C     IF(CONDV.LE.0)THEN
C        GOTO3120
C     ENDIF
C     CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
C    *LUT,P,TORAD,AS,DIF,UTL,TIME)
C     IF(LUT.GT.MAXOUT)THEN
C        B=10
C        MAXOUT=LUT
C        RB=B*1000
C        WRITE(*,*)RB
C     ENDIF
C 3120 CONDV=CONDV+F5
C     F6=FACT*CONDA
C     CONDA=CONDA+F6
C     IF(CONDA.GT.1)THEN
C        GOTO3130
C     ENDIF
C     CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
C    *LUT,P,TORAD,AS,DIF,UTL,TIME)
C     IF(LUT.GT.MAXOUT)THEN
C        B=11
C        MAXOUT=LUT
C        RB=B*1000
C        WRITE(*,*)RB
C     ENDIF
C 3130 CONDA=CONDA-2*F6
C     IF(CONDA.LE.0)THEN
```

```
C        GOTO3140
C      ENDIF
C      CALL
CPO(WIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,
C     *LUT,P,TORAD,AS,DIF,UTL,TIME)
C      IF(LUT.GT.MAXOUT)THEN
C        B=12
C        MAXOUT=LUT
C        RB=B*1000
C        WRITE(*,*)RB
C      ENDIF
C 3140 CONDA=CONDA+F6
C
C
C
      G=G+1
      IF(G.GT.200)THEN
        G=0
        GOTO3010
      ENDIF
      RB=B*1000
C
C   CURRENT STATUS OUTPUT FOR INQUISITIVE OPERATORS
C
      WRITE(*,*)RI,RB,FACT,'VARIABLE P S A W D R RL SP CA CV'
      WRITE(*,*) PREH,SUKH,ATEMP,LUT,DIAM,RATI
      WRITE(*,*) RLEN,SPRF,CONDA,CONDV
      WRITE(*,*)'PLEASE LEAVE COMPUTER ON AND DO NOT TOUCH AS
MULTI
     * VARIABLE OPTIMISATION IS IN PROGRESS'
      IF(B.EQ.0)THEN
        GOTO3010
      ENDIF
      IF(B.EQ.1.AND.GOG.EQ.16)THEN
        RATI=RATI+F1
        DIAM=DIAM-F2
        GOTO3020
      ENDIF
      IF(B.EQ.2.AND.GOG.EQ.16)THEN
        RATI=RATI-F1
        DIAM=DIAM-F2
        GOTO3020
      ENDIF
      IF(B.EQ.2.AND.GOG.EQ.15)THEN
        RATI=RATI-F1
        DIAM=DIAM+F2
        GOTO3020
```

```
      ENDIF
      IF(B.EQ.1.AND.GOG.EQ.15)THEN
        RATI=RATI+F1
        DIAM=DIAM+F2
        GOTO3020
      ENDIF
      IF(B.EQ.1.and.gog.eq.0)THEN
        RATI=RATI+F1
      GOTO3020
      ENDIF
      IF(B.EQ.2.and.gog.eq.0)THEN
        RATI=RATI-F1
        GOTO3020
      ENDIF
      IF(B.EQ.3)THEN
        DIAM=DIAM+F2
        GOTO3020
      ENDIF
      IF(B.EQ.4)THEN
        DIAM=DIAM-F2
        GOTO3020
      ENDIF
      IF(B.EQ.5)THEN
        RLEN=RLEN+F3
        GOTO3020
      ENDIF
      IF(B.EQ.6)THEN
        RLEN=RLEN-F3
        GOTO3020
      ENDIF
      IF(B.EQ.9)THEN
        CONDV=CONDV+F5
        GOTO3020
      ENDIF
      IF(B.EQ.10)THEN
        CONDV=CONDV-F5
        GOTO3020
      ENDIF
      IF(B.EQ.11)THEN
        CONDA=CONDA+F6
        GOTO3020
      ENDIF
      IF(B.EQ.12)THEN
        CONDA=CONDA-F6
        GOTO3020
      ENDIF
C
```

```
C  CALCULATE FINAL PUMPS PERFORMANCE OVER A WHOLE DAY
C
 4000 CALL COD(VOL,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,
    *CONDA,CONDV,AS,TORAD,VALOUT,WATOUT,OP,DIF,UTL,TIME)
       DFLOW=0
       DO 1820 I=1,80
         IF(TORAD(I).LE.200)THEN
           GOTO1822
         ENDIF
         VALOUT(I)=VALOUT(I)*18000/OP(I)
 1822      DFLOW=VALOUT(I)+DFLOW
 1820 CONTINUE
     RETURN
     END

     SUBROUTINE
CPOG(TORAD,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,
    *CONDV,LUT,P,DIF,UTL,TIME)
     REAL
TORAD,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,LUT,P
     REAL
CPM,CPL,CPG,MM,ML,HFG,GOLD,ONE,ZER,TOR,OMTOR,MG,WROUT,CON
     REAL EGAINT,EGAINOMT,DUM,CUN,DUMY,DIF,UTL,TIME
     DOUBLE PRECISION
TET,TEO,VALTOR,VALOMTOR,VALT1,VALOMT2,VIL
C
C  CONTROL PROGRAM FOR PUMP SIMULATION ROUTINES
INCLUDING GOLDEN SECTION
C  SEARCH TO FIND STEADY STATE OPERATING TEMPERATURE, ALSO
DEALS WITH
C  SOME OF THE FAILURE SCENARIOS. PUMPS PERFOMANCE
CALCULATED ON INPUT
C  CONDITITONS
C
C    RATI=1.67
C    RLEN=.02
C    DIAM=.24
C    CONDA=.4
C    CONDV=.002
     DUMY=0
     CUN=0
     DUM=0
     CON=0
     CPM=460
     CPL=2470
     CPG=1810
     MM=46.4
```

```
      ML=5.6
      HFG=330000
      GOLD=1/((1+(5**.5))/2)
      ONE=85
      ZER=30
      TOR=(ONE-ZER)*GOLD+ZER
C       TOR=68.5
      MG=3.5E-3*(1/(.5279-5.68E-3*TOR))
      TET=(TOR+273)*(CPM*MM+CPL*ML+CPG*MG)+HFG*MG
      CALL
WOKOUT(TET,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV
,
     *TORAD,VALTOR,WROUT,P,TOR,EGAINT,P1,DIF,UTL,TIME)
C       WRITE(*,*)'1',P1,P
      OMTOR=(ONE-ZER)*(1-GOLD)+ZER
      MG=3.5E-3*(1/(.5279-5.68E-3*OMTOR))
      TEO=(OMTOR+273)*(CPM*MM+CPL*ML+CPG*MG)+HFG*MG
      CALL
WOKOUT(TEO,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV
,
     *TORAD,VALOMTOR,WROUT,P,OMTOR,EGAINOMT,P1,DIF,UTL,TIME)
C       WRITE(*,*)'2',OMTOR
      IF(WROUT.EQ.0.AND.P.EQ.2001)THEN
        DUM=1
      ENDIF
 5210 IF(ABS(EGAINT).LE.ABS(EGAINOMT).OR.DUM.EQ.1)THEN
        DUM=0
        CUN=CUN+1
        ZER=OMTOR
        OMTOR=TOR
        TEO=TET
        EGAINOMT=EGAINT
        TOR=(ONE-ZER)*GOLD+ZER
        MG=3.5E-3*(1/(.5279-5.68E-3*TOR))
        TET=(TOR+273)*(CPM*MM+CPL*ML+CPG*MG)+HFG*MG
        CALL
WOKOUT(TET,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV
     *,TORAD,VALTOR,WROUT,P,TOR,EGAINT,P1,DIF,UTL,TIME)
C       WRITE(*,*)'3',TOR,EGAINT,P,P1
        IF(WROUT.EQ.0.AND.P.EQ.4001)THEN
          CON=CON+1
          DUM=1
          WRITE(*,*)'DEAD ROOSTER1'
          DUMY=DUMY+1
          IF(DUMY.GT.15)THEN
            WROUT=0
            GOTO5070
```

```
        ENDIF
        GOTO5220
      ENDIF
    IF(CUN.GT.15.AND.ABS(EGAINT).GT.50)THEN
      WROUT=0
      GOTO5070
    ENDIF
      IF(WROUT.EQ.0.AND.P.EQ.2001)THEN
        CON=CON+1
        DUM=1
        GOTO5210
      ENDIF
      IF(WROUT.EQ.0.AND.P.NE.2001)THEN
        CON=CON+1
        DUM=1
        GOTO5220
      ENDIF
      IF(ABS(EGAINT).LT.50)THEN
        GOTO5070
      ENDIF
      GOTO5210
    ENDIF
 5220 IF(ABS(EGAINT).GT.ABS(EGAINOMT).OR.DUM.EQ.1)THEN
      DUM=0
      CUN=CUN+1
      ONE=TOR
      TOR=OMTOR
      TET=TEO
      EGAINT=EGAINOMT
      OMTOR=(ONE-ZER)*(1-GOLD)+ZER
      MG=3.5E-3*(1/(.5279-5.68E-3*OMTOR))
      TEO=(OMTOR+273)*(CPM*MM+CPL*ML+CPG*MG)+HFG*MG
      CALL
WOKOUT(TEO,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,

*CONDV,TORAD,VALOMTOR,WROUT,P,OMTOR,EGAINOMT,P1,DIF,UTL,TI
ME)
C     WRITE(*,*)'4',OMTOR,EGAINOMT,P,P1
    ENDIF
    IF(CUN.GT.15.AND.ABS(EGAINOMT).GT.50)THEN
      WROUT=0
      GOTO5070
    ENDIF
    IF(CON.EQ.CUN.AND.CON.GT.10)THEN
      WROUT=0
      GOTO5070
    ENDIF
```

```
      IF(ABS(EGAINOMT).LE.50)THEN
        GOTO5070
      ENDIF
        IF(WROUT.EQ.0.AND.P.EQ.4001)THEN
          CON=CON+1
          DUM=1
          WRITE(*,*)'DEAD ROOSTER2'
          DUMY=DUMY+1
          IF(DUMY.GT.15)THEN
            WROUT=0
            GOTO5070
          ENDIF
          GOTO5210
        ENDIF
      IF(WROUT.EQ.0)THEN
        CON=CON+1
        DUM=1
        GOTO5210
      ENDIF
      GOTO5210
 5070 LUT=WROUT
      RETURN
      END

      SUBROUTINE
CPO(VOL,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,
     *CONDV,LUT,P,TORAD,AS,DIF,UTL,TIME)
      REAL
TORAD(80),PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV
      REAL
CPM,CPL,CPG,MM,ML,HFG,GOLD,ONE,ZER,TOR,OMTOR,MG,WROUT,CON
      REAL
EGAINT,EGAINOMT,DUM,CUN,DUMY,VOL,LUT,P,AS(80),WATOUT
      REAL VALOUT(80),OP(80),T1,T2,T3,DIF,UTL,TIME
      DOUBLE PRECISION
TET,TEO,VALTOR,VALOMTOR,VALT1,VALOMT2,VIL
C
C   CONTROL PROGRAM FOR PUMP SIMULATION ROUTINES
INCLUDING GOLDEN SECTION
C   SEARCH TO FIND STEADY STATE OPERATING TEMPERATURE, ALSO
DEALS WITH
C   SOME OF THE FAILURE SCENARIOS. PUMPS PERFOMANCE
CALCULATED OVER A
C   WHOLE DAYS INPUT CONDITIONS
C
C      RATI=1.71
C      RLEN=.02
```

```
C     DIAM=.49
C     CONDA=.4
C     CONDV=.002
C     VOL=800
      DUMY=0
      CUN=0
      DUM=0
      CON=0
      CPM=460
      CPL=2470
      CPG=1810
      MM=46.4
      ML=5.6
      HFG=330000
      GOLD=1/((1+(5**.5))/2)
      ONE=85
      ZER=30
      TOR=(ONE-ZER)*GOLD+ZER
      TOR=62.44
      MG=3.5E-3*(1/(.5279-5.68E-3*TOR))
      TET=(TOR+273)*(CPM*MM+CPL*ML+CPG*MG)+HFG*MG
      CALL
WOKOUT(TET,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV
,
     *VOL,VALTOR,WROUT,P,TOR,EGAINT,P1,DIF,UTL,TIME)
C     WRITE(*,*)'1',P1,P
      OMTOR=(ONE-ZER)*(1-GOLD)+ZER
      MG=3.5E-3*(1/(.5279-5.68E-3*OMTOR))
      TEO=(OMTOR+273)*(CPM*MM+CPL*ML+CPG*MG)+HFG*MG
      CALL
WOKOUT(TEO,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV
,
     *VOL,VALOMTOR,WROUT,P,OMTOR,EGAINOMT,P1,DIF,UTL,TIME)
C     WRITE(*,*)'2',OMTOR
      IF(WROUT.EQ.0.AND.P.EQ.2001)THEN
        DUM=1
      ENDIF
 5210 IF(ABS(EGAINT).LE.ABS(EGAINOMT).OR.DUM.EQ.1)THEN
        DUM=0
        CUN=CUN+1
        ZER=OMTOR
        OMTOR=TOR
        TEO=TET
        EGAINOMT=EGAINT
        TOR=(ONE-ZER)*GOLD+ZER
        MG=3.5E-3*(1/(.5279-5.68E-3*TOR))
        TET=(TOR+273)*(CPM*MM+CPL*ML+CPG*MG)+HFG*MG
```

```
      CALL
WOKOUT(TET,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV
    *,VOL,VALTOR,WROUT,P,TOR,EGAINT,P1,DIF,UTL,TIME)
C     WRITE(*,*)'3',TOR,EGAINT,P,P1
      IF(WROUT.EQ.0.AND.P.EQ.4001)THEN
        CON=CON+1
        DUM=1
        WRITE(*,*)'DEAD ROOSTER1'
        DUMY=DUMY+1
        IF(DUMY.GT.15)THEN
          WROUT=0
          GOTO5070
        ENDIF
        GOTO5220
      ENDIF
    IF(CUN.GT.15.AND.ABS(EGAINT).GT.50)THEN
      WROUT=0
      GOTO5070
    ENDIF
      IF(WROUT.EQ.0.AND.P.EQ.2001)THEN
        CON=CON+1
        DUM=1
        GOTO5210
      ENDIF
      IF(WROUT.EQ.0.AND.P.NE.2001)THEN
        CON=CON+1
        DUM=1
        GOTO5220
      ENDIF
      IF(ABS(EGAINT).LT.50)THEN
        CALL CODG(VOL,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,
    *CONDA,CONDV,AS,TORAD,VALOUT,WATOUT,OP,DIF,UTL,TIME)
        WROUT=0
        DO 1820 I=1,80,2
          IF(TORAD(I).LE.200)THEN
            GOTO1822
          ENDIF
          VALOUT(I)=VALOUT(I)*2*18000/OP(I)
1822        WROUT=VALOUT(I)+WROUT
1820      CONTINUE
        GOTO5070
      ENDIF
      GOTO5210
    ENDIF
5220 IF(ABS(EGAINT).GT.ABS(EGAINOMT).OR.DUM.EQ.1)THEN
      DUM=0
      CUN=CUN+1
```

```
          ONE=TOR
          TOR=OMTOR
          TET=TEO
          EGAINT=EGAINOMT
          OMTOR=(ONE-ZER)*(1-GOLD)+ZER
          MG=3.5E-3*(1/(.5279-5.68E-3*OMTOR))
          TEO=(OMTOR+273)*(CPM*MM+CPL*ML+CPG*MG)+HFG*MG
          CALL
WOKOUT(TEO,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,

*CONDV,VOL,VALOMTOR,WROUT,P,OMTOR,EGAINOMT,P1,DIF,UTL,TIME
)
C         WRITE(*,*)'4',OMTOR,EGAINOMT,P,P1
        ENDIF
        IF(CUN.GT.15.AND.ABS(EGAINOMT).GT.50)THEN
          WROUT=0
          GOTO5070
        ENDIF
        IF(CON.EQ.CUN.AND.CON.GT.10)THEN
          WROUT=0
          GOTO5070
        ENDIF
        IF(ABS(EGAINOMT).LE.50)THEN
          CALL CODG(VOL,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,
     *CONDA,CONDV,AS,TORAD,VALOUT,WATOUT,OP,DIF,UTL,TIME)
          WROUT=0
          DO 1830 I=1,80,2
            IF(TORAD(I).LE.200)THEN
              GOTO1832
            ENDIF
            VALOUT(I)=VALOUT(I)*2*18000/OP(I)
 1832       WROUT=VALOUT(I)+WROUT
 1830     CONTINUE
          GOTO5070
        ENDIF
          IF(WROUT.EQ.0.AND.P.EQ.4001)THEN
            CON=CON+1
            DUM=1
            WRITE(*,*)'DEAD ROOSTER2'
            DUMY=DUMY+1
            IF(DUMY.GT.15)THEN
              WROUT=0
              GOTO5070
            ENDIF
            GOTO5210
          ENDIF
        IF(WROUT.EQ.0)THEN
```

```
      CON=CON+1
      DUM=1
      GOTO5210
    ENDIF
    GOTO5210
 5070 LUT=WROUT
    RETURN
    END

    SUBROUTINE
WOKOUT(INTEIN,PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,

*CONDA,CONDV,TORAD,INTEOU,WROUT,P,STEMP,EGAIN,P1,DIF,UTL,TIM
E)
    REAL
PREH,SUKH,DIAM,RATI,RLEN,ATEMP,SPRF,CONDA,CONDV,LMT
    REAL TORAD,WROUT,P,STEMP,DEFL1,DEFL2,DEFL3,DEFL4,MAXTRA
    REAL
HD,PI,CONDT,PD,TK,TC,PUMVOL,NMG,ENERGY,PREE,ATMOS,HC1
    REAL
CPM,CPL,CPG,MM,ML,HFG,MG,MINGA1,RADIU1,RADIU2,RADIU3,RADIU4
    REAL
ANGLE1,ANGLE2,ANGLE3,ANGLE4,SECAR1,SECAR2,SECAR3,SECAR4
    REAL
SEGAR1,SEGAR2,SEGAR3,SEGAR4,SEGVO1,SEGVO2,SEGVO3,SEGVO4
    REAL
TOTVOL,ARE1,ARE2,INITRA,MXTG,VOLINC,U1SQ,U2SQ,U1,U2,PW,G
    REAL
FLUVOL,TRA,VOL,INCR,PUGAVO,GASVOL,MASS,P1,EGAIN,GAIN
    REAL
PL,PV,D,UL,CONRAT,MASSCH,COLLTI,RAT,SP,K,ACCEL,GASM,DRAG
    REAL
TIME,TIME2,TIME3,OFF,PLTEMP1,TLOW,TER,DIF,UTL,DOOG,DRIG
    DOUBLE PRECISION
INTEIN,INTEOU,INTUN,NETOTE,TOTVAL,CONVAL,HC
C
C   PUMP-PRIME MOVER-COLLECTOR SIMULATION
C
    INTUN=INTEIN
    HD=DIAM/2
    PI=3.141592654
    EGAIN=0
    CONDT=290
    CPM=460
    CPL=2470
    CPG=1810
    MM=46.4
```

```
      MG=3.5E-3*(1/(0.5279-5.68E-3*STEMP))
      ML=5.6
      HFG=330000
      ATMOS=10
      PD=DIAM*PI
      ARE1=PI*HD**2
      ARE2=PI*((RATI*HD)**2-HD**2)
C
C  VALVE OPEN TO COLLECTOR, WORK OUT PRESSURE OF VAPOUR
WHEN THE VOLUMES OF
C  OF COLLECTOR AND PUMP ARE JOINED
C
      TK=STEMP+273
      PUMVOL=.001*PI*HD**2+PD*1E-4
      NMG=PUMVOL*1/(.5279-5.68E-3*(CONDT-273))
      MG=MG+NMG
      ENERGY=NMG*(CONDT*CPG+HFG)
      EGAIN=ENERGY
      NETOTE=INTEIN+ENERGY

TK=(NETOTE/(CPM*MM+CPL*ML+CPG*MG)-(HFG*MG/(CPM*MM+CPL*
ML+CPG*MG)))
      PREE=133.322*10**(6.8643-1070.62/(232.696+TK-273))
C
C  DEFLECTION OF DIAPHRAGM CALCS
C

DEFL1=RLEN*(3*ABS(PREE-(ATMOS-SUKH)*1E4)*RLEN/7.186E5)**.333333
      MINGA1=2*RLEN/PI
      MAXTRA=2*((RLEN**2)-(MINGA1**2))**.5
      RADIU1=(DEFL1**2+RLEN**2/4)/(2*DEFL1)
      ANGLE1=2*ASIN((RLEN/2)/RADIU1)
      SECAR1=PI*RADIU1**2*ANGLE1/(2*PI)
      SEGAR1=SECAR1-RADIU1*RLEN/2+DEFL1*RLEN/2
      DIAR=2*PI*HD
      SEGVO1=DIAR*SEGAR1
C

DEFL2=RLEN*(3*(ATMOS-(ATMOS-SUKH))*1E4*RLEN/7.186E5)**.333333
      RADIU2=(DEFL2**2+RLEN**2/4)/(2*DEFL2)
      ANGLE2=2*ASIN((RLEN/2)/RADIU2)
      SECAR2=PI*RADIU2**2*ANGLE2/(2*PI)
      SEGAR2=SECAR2-RADIU2*RLEN/2+DEFL2*RLEN/2
      SEGVO2=DIAR*RATI*SEGAR2
      CONDP=133.322*10**(6.8643-1070.62/(232.696+CONDT-273))
C
      DEFL3=RLEN*(3*((PREH+10)*1E4-CONDP)*RLEN/7.168E5)**.333333
```

```
      RADIU3=(DEFL3**2+RLEN**2/4)/(2*DEFL3)
      ANGLE3=2*ASIN((RLEN/2)/RADIU3)
      SECAR3=PI*RADIU3**2*ANGLE3/(2*PI)
      SEGAR3=SECAR3-RADIU3*RLEN/2+DEFL3*RLEN/2
      SEGVO3=DIAR*SEGAR3
C
      DEFL4=RLEN*(3*PREH*1E4*RLEN/7.168E5)**.333333
      RADIU4=(DEFL4**2+RLEN**2/4)/(2*DEFL4)
      ANGLE4=2*ASIN((RLEN/2)/RADIU4)
      SECAR4=PI*RADIU4**2*ANGLE4/(2*PI)
      SEGAR4=SECAR4-RADIU4*RLEN/2+DEFL4*RLEN/2
      SEGVO4=DIAR*RATI*SEGAR4
C
      TOTVOL=SEGVO1+SEGVO2+SEGVO3+SEGVO4
C
C  WORK OUT PRESSURE AFTER TRAVEL TO TAKE UP DEFLECTION OF
DIAPHRAGM
C
      INITRA=TOTVOL/ARE2
      VOLINC=INITRA*ARE1
      MXTG=VOLINC*1/(.5279-5.68E-3*(TK-273))
      ML=ML-MXTG
      MG=MG+MXTG

TK=(NETOTE/(CPM*MM+CPL*ML+CPG*MG)-(HFG*MG/(CPM*MM+CPL*
ML+CPG*MG)))
      PREE=133.322*10**(6.8643-1070.62/(232.696+TK-273))
      P=0
      U1SQ=0
      G=9.81
      PW=1000
      U1=0
      TRAVEL=INITRA
      VOL=VOLINC
      ACCPRE=0
C
C  RUN THROUGH PUMP TRAVEL AND PRESSURE CALCS FOREACH
1/20 SECOND INTERVAL
C
 6000 IF(TRAVEL.LT.MAXTRA)THEN
         P=P+1

ACCPRE=(PI*(RATI*HD)**2*1E4*ATMOS-ARE2*(((ATMOS-SUKH)*1E4)-
     *ACCPRE)-ARE1*PREE)/ARE2
C  ALLOWANCE FOR CAVITATION
         IF((SUKH-ACCPRE/1E4).GT.9.5)THEN
            ACCPRE=(SUKH-9.5)*1E4
```

```
      ENDIF
C     WRITE(*,*)'1',ACCPRE,ACCPREE
      GAIN=1/20*(((.6625-3.98*(TK-ATEMP-273)/TORAD)*2.9*TORAD-110*
     *(TK-ATEMP-273)*PD*RLEN-2.5*(TK-ATEMP-273)*PI*HD**2*2.2))
      NETOTE=NETOTE+GAIN

TK=(NETOTE/(CPM*MM+CPL*ML+CPG*MG)-(HFG*MG/(CPM*MM+CPL*
ML+CPG
     **MG)))
      EGAIN=EGAIN+GAIN
      PREE=133.322*10**(6.8643-1070.62/(232.696-273+TK))
C   FAILURE SCENARIO SIGNALS
      IF(P.GT.2000)THEN
        GOTO 6070
      ENDIF
      IF(ACCPRE.GE.0)THEN
        GOTO 6000
      ENDIF
C
C   WORK OUT WATER VELOCITY AND VOLUME OF FLUID
TRANSPOSED
C
      U1SQ=U1**2
      ACCEL=(ACCPRE*(-1)-10000*U1SQ)/8000
      U2=ACCEL/20+U1
      IF(U2.LT.0)THEN
        U2=0
      ENDIF
      FLUVOL=(U1+U2)*2.85E-5
      U1=U2*(7-U2/20)/7
      TRA=FLUVOL/ARE2
      TRAVEL=TRA+TRAVEL
      VOL=VOL+TRA*ARE1
    ELSE
C
C   FINAL INCREMENT OF TIME < 1/20 SECOND
C
      INCR=(MAXTRA-(TRAVEL-TRA))/TRA
      VOL=(INCR-1)*TRA*ARE1+VOL
      TRAVEL=TRAVEL+TRA*(INCR-1)
      P=P+INCR
      NETOTE=NETOTE+(INCR-1)*GAIN
      EGAIN=EGAIN+(INCR-1)*GAIN
      GOTO6010
    ENDIF
    GOTO6000
 6010 P1=P
```

```
C
C   VALVE CHANGE OVER TO THE CONDENSER
C   ADD VOLUME OF CONDENSER TO PUMPS VAPOR CHAMBER AND
WORK OUT PRESSURE
C
      PGC=1/(.5279-5.68E-3*(CONDT-273))
      CONVAL=CONDV*PGC*(CONDT*CPG+HFG)
      PUGAVO=MAXTRA*ARE1+SEGVO1
      PGP=1/(.5279-5.68E-3*(TK-273))
      TOTVAL=CONVAL+PUGAVO*PGP*(CPG*TK+HFG)
      GASVOL=CONDV+PUGAVO
      PG=(PGC*CONDV/GASVOL)+(PGP*PUGAVO/GASVOL)
      TC=TOTVAL/(GASVOL*PG*CPG)-HFG/CPG
      INITRA=TOTVOL/ARE2
      VOLINC=GASVOL-INITRA*ARE1
      PREE=133.322*10**(6.8643-1070.62/(232.696-273+TC))
      U1SQ=0
      U1=0
C   START CONDENSING ROUTINE FOR 1/20 SECOND INTERVALS
C   PUMP TRAVEL AND PRESSURE CALCS.
      TRAVEL=INITRA
      VOL=VOLINC
      MASS=0
      GASMAS=GASVOL*PG
      GASM=GASMAS
      ACCPRE=0
 6020 IF(TRAVEL.LT.MAXTRA)THEN
         P=P+1

ACCPRE=(PI*(RATI*HD)**2*ATMOS*1E4-ARE2*(((PREH+ATMOS)*1E4)+
     *ACCPRE)-ARE1*PREE-SPRF)/ARE2
         PL=600
         PV=1/(.5279-5.68E-3*(TK-273))
         HFG=394800-1350*(TK-273)
         K=0.1
         D=0.02
         UL=1.65E-4
         HC1=0.725*((PL*(PL-PV)*G*HFG*K**3)/(D*UL*(TK-CONDT)))**0.25
         HC=1/(1/HC1+.002/50+1/1000)
         LMT=((TK-TC)-1.4)/LOG((TK-TC)/1.4)
        CONRAT=CONDA*LMT*HC+110*(TC-ATEMP-273)*PD*RLEN+2.5*(TC-
     *(ATEMP+273))*PI*HD**2*2.2
         HFG=330000
         MASSCH=CONRAT/(HFG*20)
         MASS=MASS+MASSCH
         IF(MASS.GT.GASM)THEN
            MASS=GASM
```

```
         MASSCH=0
       ENDIF
       PG=1/(.5279-5.68E-3*(TC-273))
       GASMAS=GASMAS-MASSCH
       TOTVAL=TOTVAL-CONRAT/20

TC=TOTVAL/(CPG*GASMAS+MASS*CPL)-HFG*GASMAS/(CPG*GASMAS+
MASS*CPL)
       IF(TC.LT.290)THEN
         TC=290
       ENDIF
       PREE=133.322*10**(6.8643-1070.62/(232.696-273+TC))
       IF((P-P1).GT.2000)THEN
         P=4001
         GOTO 6070
       ENDIF
       IF(ACCPRE.LE.0)THEN
         GOTO 6020
       ENDIF
       U1SQ=U1**2
       ACCEL=(ACCPRE-6000*U1SQ)/8000
       U2=ACCEL/20+U1
       IF(U2.LT.0)THEN
         U2=0
       ENDIF
       FLUVOL=(U1+U2)*2.85E-5
       U1=U2*(1.7-U2/20)/1.7
C      WRITE(*,*)'2',U1,U2,PREE
       TRA=FLUVOL/ARE2
       TRAVEL=TRA+TRAVEL
       VOL=VOL-TRA*ARE1
     ELSE
C
C   FINAL INCREMENT OF TIME < 1/20 SECOND
C
       INCR=(MAXTRA-(TRAVEL-TRA))/TRA
       VOL=VOL-(INCR-1)*TRA*ARE1
       TRAVEL=TRAVEL+TRA*(INCR-1)
       P=P+INCR
       MASS=MASS+MASSCH*(INCR-1)
       GOTO6030
     ENDIF
     GOTO6020
 6030 COLLTI=(P-P1)/20
C
C   CALCULATE COLLECTOR ENERGY GIVEN CONDENSING TIME
C
```

```
      PG=1/(.5279-5.68E-3*(TK-273))
      EGAIN=EGAIN-((ARE1*MAXTRA+SEGVO1+PUMVOL)*PG*HFG)

EGAIN=EGAIN+COLLTI*TORAD*2.9*(.6625-3.98*(TK-ATEMP-273)/TORAD)
      WROUT=MAXTRA*ARE2-TOTVOL
      GOTO6090
 6070 WROUT=0
 6090 WROUT=WROUT*1
      TER=CPM*MM+CPL*ML+CPG*MG
C
C  EQUATE DOWN TIME DUE TO COOLING OF COLLECTOR DURING
CLOUDY PERIOD
C
      DRIG=EXP((-1)*UTL*TIME*60/(TER/2.9))
      DRAG=DIF*TORAD-UTL*(STEMP-ATEMP)
      PLTEMP1=ATEMP+((-1)/UTL)*((-1)*DIF*TORAD+(DRIG*DRAG))
C       WRITE(*,*)'1',ATEMP,TIME,PLTEMP1,STEMP,DRIG,DRAG
      IF(PLTEMP1.GE.STEMP)THEN
        OFF=TIME/15
        GOTO132
      ENDIF

DOOG=((TORAD-UTL*(STEMP-ATEMP))/(TORAD-UTL*(PLTEMP1-ATEMP)
))
C       WRITE(*,*)'2',ATEMP,TIME,PLTEMP1,STEMP,DOOG
      IF(DOOG.LE.0)THEN
        OFF=1
        GOTO132
      ENDIF
      TIME2=TER*(-1)*LOG(DOOG)/(UTL*2.9)
      TIME3=TIME*60+TIME2
      OFF=TIME3/(15*60)
C       WRITE(*,*)'3',ATEMP,TIME,PLTEMP1,STEMP,TIME2,OFF
C  FIND OUTPUT CORRECTED FOR DOWN TIME
C
      OFF=0
  132 WROUT=WROUT*(1-OFF)
      RETURN
      END

*$include SETSCR.FOR
*$include SUNAG.FOR
*$include COEFF.FOR
*$include GLASS.FOR
*$include AMBIENT.FOR
*$include TOLOSS.FOR
```

```
      SUBROUTINE
AMBIENT(HOUR,DAYNUM,MONNUM,LSIGN,DAYDIF,SEADIF,MAXT,
     *TAMB,LATDEG)
      INTEGER DAYNUM,MONNUM,LSIGN
      REAL HOUR,DAYDIF,SEADIF,MAXT,TAMB,LATDEG,MMEAN,TSEA,PI
      PI=3.141592654
C
C  DAILY VARIATIONS DUE TO THE SEASONS (IGNORING RAINY
SEASONS)
C
      MMEAN=MAXT-SEADIF/2-DAYDIF/2
C
C  FOR N + S AND LAT GE 23.5 DEGREES
      IF(LSIGN.EQ.-1.AND.LATDEG.GE.23.5) THEN
        TSEA=MMEAN+SEADIF*(COS((DAYNUM-11)*2*PI)/265)/2
        GOTO50
      ENDIF
      IF(LSIGN.EQ.1.AND.LATDEG.GE.23.5) THEN
        TSEA=MMEAN-SEADIF*(COS((DAYNUM-11)*2*PI)/265)/2
        GOTO50
      ENDIF
C
C  FOR N + S AND LAT LT 23.5 DEG
C
      PVESIZ=.5*LATDEG/23.5
      NVESIZ=PVESIZ-1
      PVEHEI=PVESIZ*SEADIF
      DAYSPV=182*LATDEG/23.5+182
      DAYSNV=365-DAYSPV
      IF(LSIGN.EQ.1)THEN
        IF(DAYNUM.LT.182)THEN
          DAYS=182+DAYNUM
        ENDIF
        IF(DAYNUM.GE.182)THEN
          DAYS=DAYNUM-182
        ENDIF
      ENDIF
      IF(LSIGN.EQ.-1)THEN
        DAYS=DAYNUM
      ENDIF
      IF(DAYS.LE.(DAYSPV/2))THEN

TSEA=MMEAN+(SEADIF-(PVEHEI/2)+(PVEHEI/2)*(COS(DAYS/(DAYSPV/
2))
     **PI+PI))/2
      ENDIF
      IF((365-DAYS).LE.(DAYSPV/2))THEN
```

```
      TSEA=MMEAN+(SEADIF-(PVEHEI/2)+(PVEHEI/2)*(COS((DAYS-365)/
     *(DAYSPV/2))*PI+PI))/2
      ENDIF
      IF(DAYS.GT.(DAYSPV/2).AND.DAYS.LT.(365-DAYSPV/2))THEN

TSEA=MMEAN+SEADIF*COS((DAYS-DAYSPV/4)*2*PI/(DAYSNV+DAYSPV
/2))
     */2
      ENDIF
C
C  NOW ADD THE HOURLY VARIATION TO THE DAILY MEAN
C
   50 IF(HOUR.LT.6) THEN
        TAMB=TSEA+DAYDIF*COS(PI-((6-HOUR)*PI/16))/2
      ENDIF
      IF(HOUR.GE.6.AND.HOUR.LE.14) THEN
        TAMB=TSEA+DAYDIF*COS(((HOUR-6)*PI/8)-PI)/2
      ENDIF
      IF(HOUR.GT.14) THEN
        TAMB=TSEA+DAYDIF*COS((HOUR-14)*PI/16)/2
      ENDIF
      RETURN
      END


      SUBROUTINE COEFF(MONTH,A,B,E)
C
C   GIVES COEFFICIENTS FOR THE EQUATIONS USED IN THE MAIN
PROGRAMME
C
      REAL A,B,E,MONTH
      IF(MONTH.EQ.0) GOTO130
      IF(MONTH.EQ.1) GOTO20
      IF(MONTH.EQ.2) GOTO30
      IF(MONTH.EQ.3) GOTO40
      IF(MONTH.EQ.4) GOTO50
      IF(MONTH.EQ.5) GOTO60
      IF(MONTH.EQ.6) GOTO70
      IF(MONTH.EQ.7) GOTO80
      IF(MONTH.EQ.8) GOTO90
      IF(MONTH.EQ.9) GOTO100
      IF(MONTH.EQ.10) GOTO110
      IF(MONTH.EQ.11) GOTO120
      IF(MONTH.EQ.12) GOTO130
      IF(MONTH.EQ.13) GOTO20
   20 A=1230
      B=.142
```

```
   E=.058
   GOTO140
30 A=1215
   B=.144
   E=.060
   GOTO140
40 A=1186
   B=.156
   E=.071
   GOTO140
50 A=1136
   B=.180
   E=.097
   GOTO140
60 A=1104
   B=.196
   E=.121
   GOTO140
70 A=1088
   B=.205
   E=.134
   GOTO140
80 A=1085
   B=.207
   E=.136
   GOTO140
90 A=1107
   B=.201
   E=.122
   GOTO140
100 A=1151
   B=.177
   E=.092
   GOTO140
110 A=1192
   B=.160
   E=.073
   GOTO140
120 A=1221
   B=.149
   E=.063
   GOTO140
130 A=1233
   B=.142
   E=.057
   GOTO140
140 RETURN
```

```
      END


      SUBROUTINE CURPOS(PAGE,ROW,COLUMN)
C
C  THIS SUBROUTINE WILL POSITION THE CURSOR USING
C  SOFTWARE INTERRUPTS.
C
C  DEFINE REGISTERS: THESE CORRESPOND TO THE ELEMENT OF AN
C  ARRAY WHICH IS TO CONTAIN THE VALUES OF THE REGISTERS.
C
      IMPLICIT INTEGER (A-Z)
      INTEGER*2 AX,BX,CX,DX,BP,DI,SI,ES,DS,FLAGS
      PARAMETER (AX=1,BX=2,CX=3,DX=4,BP=5,DI=6,SI=7)
      PARAMETER (ES=8,DS=9,FLAGS=10)
      INTEGER*2 REGS(10)
C
      AH = 2
      AL = 0
      BH = PAGE
      BL = 0
      DH = ROW
      DL = COLUMN
      REGS(AX) = AH * 256 + AL
      REGS(BX) = BH * 256 + BL
      REGS(DX) = DH * 256 + DL
      CALL INTR(16,REGS)
      RETURN
      END


      SUBROUTINE
GLASS(N,TRANS,TRINT,SORB,HERP,THICK,ABCOEF,REFG)
C
C  THIS FINDS THE SOLAR RADIATION TRANSMITTED TO A ONE
METER SQUARE
C    SURFACE BEHIND A CERTAIN AMONT OF GLASS AT A CERTAIN
ANGLE (SPECIFIRED)
C
      REAL IDASH,RP,RS,TRINT,TRANS,SORB
C
C  NON REFLECTED COMPONENT
C
      IDASH=ASIN(SIN(HERP)/REFG)
      RP=((TAN(HERP-IDASH))**2)/(((TAN(HERP+IDASH))**2)*2)
      RS=((SIN(HERP-IDASH))**2)/(((SIN(HERP+IDASH))**2)*2)
      TRINT=.5*(((1-RP)/(1+(2*N-1)*RP))+((1-RS)/(1+(2*N-1)*RS)))
```

```
C
C   ABSORPTANCE COMPONENT
C
      TRANS=TRINT*((1-ABCOEF)**N)
      SORB=TRANS-TRINT
      RETURN
      END


      SUBROUTINE SCROLL(AH,AL,CH,CL,DH,DL,BH)
C
C   THIS SUBROUTINE DEFINES A "WINDOW" AND CONTROLS THE
SCROLLING OF
C   TEXT WITHIN THAT WINDOW.  IT WILL CONTROL THE NUMBER OF
ROWS THE
C   SCREEN SCROLLS AND CONTROLS THE ATTRIBUTES OF THE ROWS.
C
C   THE FOLLOWING LINES DESCRIBE THE SUBROUTINE'S ARGUMENTS.
C
C       AH -- CONTROLS WHETHER THE SCREEN SCROLLS UP OR DOWN
C               AH = 6 MAKES WINDOW SCROLL UP
C               AH = 7 MAKES WINDOW SCROLL DOWN
C       AL -- NUMBER OF INPUT LINES BLANKED AT SCROLLING
C       CH -- TOP ROW OF WINDOW
C       CL -- LEFTMOST COLUMN OF WINDOW
C       DH -- BOTTOM ROW OF WINDOW
C       DL -- RIGHTMOST COLUMN OF WINDOW
C       BH -- CONTROLS THE ATTRIBUTES ASSIGNED TO THE INPUT
LINES
C               BH = 0 GIVES BLANK LINES
C               BH = 1 GIVES UNDERLINED LINES
C
C   DEFINE REGISTERS: THESE CORRESPOND TO THE ELEMENT OF AN
C   ARRAY WHICH IS TO CONTAIN THE VALUES OF THE REGISTERS.
      IMPLICIT INTEGER (A-Z)
      INTEGER*2 AX,BX,CX,DX,BP,DI,SI,ES,DS,FLAGS
      PARAMETER (AX=1,BX=2,CX=3,DX=4,BP=5,DI=6,SI=7)
      PARAMETER (ES=8,DS=9,FLAGS=10)
      INTEGER*2 REGS(10)
C
      REGS(AX) = AH * 256 + AL
      REGS(BX) = BH * 256
      REGS(CX) = CH * 256 + CL
      REGS(DX) = DH * 256 + DL
C
      CALL INTR(16,REGS)
      RETURN
```

```
      END


      SUBROUTINE SETSCR(I)
C
C   THIS WILL SET THE SCREEN TO THE FOLLOWING MODES:
C     0 = 40 x 25 MONOCHROME
C     1 = 40 x 25 COLOR
C     2 = 80 x 25 MONOCHROME
C     3 = 80 x 25 COLOR
C     4 = 320 x 200 COLOR
C     5 = 320 x 200 MONOCHROME
C     6 = 640 x 200 MONOCHROME
C     7 = ENABLE WORD WRAP
C
      INTEGER I
      CHARACTER A
      A = CHAR(I+48)
      WRITE(*,*) '','[','=',A,'h'
      END


      SUBROUTINE
SUNAG(ORIENT,LATRAD,MONTH,DAY,HOUR,BETA,GAMMA,U,V,W)
      INTEGER MONTH,DAY,TOTOMN(12)
      REAL
DAYOFY,HOUR,LATRAD,ALT,AZI,LSTRAD,DECRAD,BETA,GAMMA,PI,ORI
ENT
      REAL U,V,W,ROOT,DUM,DOM
      PARAMETER ( PI = 3.141592654 )
C
C   THIS PROGRAM WILL DETERMINE THE ALTITUDE AND AZIMUTH
ANGLES
C   OF THE SUN AT ANY POINT ON THE EARTH AT ANY TIME OF THE
YEAR.
C
C          REQUIRED ARGUMENTS:
C             LATRAD -- LATITUDE ANGLE IN RADIANS.  THE
NORTHERN
C                 HEMISPHERE ANGLES ARE POSITIVE AND THE
SOUTHERN
C                 HEMISPHERE ANGLES ARE NEGATIVE.  THE ABSOLUTE
C                 VALUE OF THE LATITUDE MUST BE LESS THAN OR
EQUAL
C                 TO PI/2.
C             MONTH -- THE VALID RANGE OF INPUTS ARE 1 THROUGH
12.
```

```
C              DAY -- THE VALID RANGE CORRESPONDING TO EACH
MONTH.
C                 NOTE THAT 29 FEBRUARY IS NOT AN OPTION.
C              LST -- LOCAL STANDARD TIME.  THE VALID RANGES ARE
C                 BETWEEN 00.00 AND 24.00.  NOTE THAT FRACTIONS
C                 OF HOURS ARE TO BE EXPRESSED IN DECIMAL FORM.
C
C           RETURNED ARGUMENTS:
C              ALT -- THE ALTITUDE ANGLE IN RADIANS MEASURED
FROM
C                 THE LOCAL HORIZON.
C              AZI -- THE AZIMUTH ANGLE IN RADIANS MEASURED
C                 CLOCKWISE FROM TRUE NORTH.
C
C
C
C   THIS IS A LIST OF THE NUMBER OF DAYS UNTIL A GIVEN MONTH
C
      TOTOMN(1)=0
      TOTOMN(2)=31
      TOTOMN(3)=59
      TOTOMN(4)=90
      TOTOMN(5)=120
      TOTOMN(6)=151
      TOTOMN(7)=181
      TOTOMN(8)=212
      TOTOMN(9)=243
      TOTOMN(10)=273
      TOTOMN(11)=304
      TOTOMN(12)=334
C
C   FIND DAY OF THE YEAR
C
      DAYOFY=TOTOMN(MONTH)+DAY
C
C   CALCULATE THE AZIMUTH AND ALTITUDE ANGLES
C
      LSTRAD = (HOUR-12.0)*PI/12.0
C
      DECRAD = SIN((DAYOFY-81.5)*2*PI/365.0)*0.410152
C

DUM=(COS(LATRAD)*COS(LSTRAD)*COS(DECRAD)+SIN(LATRAD)*SIN(D
ECRAD))
      IF(DUM.GT.1)THEN
        DUM=1
        WRITE(*,*)'SUNAG ALMOST AT ERROR'
```

```
      ENDIF
      BETA = ASIN(DUM)
C
      DOM=(SEC(BETA)*(COS(LATRAD)*SIN(DECRAD)-COS(DECRAD)*SIN(
     *LATRAD)*COS(LSTRAD)))
      IF(DOM.GT.1)THEN
        WRITE(*,*)'SUNAG ALMOST AT ERROR'
        DOM=1
      ENDIF
      GAMMA = ACOS(DOM)
C
C  THIS CORRECTS FOR THE FACT THAT THE EQUATIONS RETURN
THE AZIMUTH
C  ANGLES MEASURED CLOCKWISE FROM TRUE NORTH BETWEEN
SOLAR NOON AND
C  SOLAR MIDNIGHT.
C
      IF(LSTRAD.GT.0.0) THEN
        GAMMA = 2*PI-GAMMA
      ENDIF
C
      ALT = BETA
      AZI = GAMMA
      IF(AZI.LE.0)THEN
        WRITE(*,*)'SUNAG ALMOST AT ERROR'
        GOTO 1234
      ENDIF
C
      AZI=AZI+INT((ABS(AZI)-AZI)/2/AZI+.1)*2*PI
C
 1234 U=-COS(AZI)*COS(ALT)
      V=-SIN(AZI)*COS(ALT)
      W=-SIN(ALT)
C
      RETURN
      END
C
      REAL FUNCTION SEC(BETA)
C
C  THIS RETURNS THE SECANT OF THE ANGLE BETA
C  IF THE ANGLE IS PI/2 OR 3*PI/2 THEN A VAL OF 64000.0
C  IS RETURNED.
C
      PARAMETER(PI=3.1415927)
      IF(BETA.EQ.(PI/2.0).OR.BETA.EQ.(3.0*PI/2.0)) THEN
        SEC = 64000.0
      ELSE
```

```
      SEC = 1.0/COS(BETA)
    ENDIF
    RETURN
    END


    SUBROUTINE
TOLOSS(TA,SPACE,EMITP,N,OUTCON,EMITG,PTEMP,COLALTR,
   *UTL)
    INTEGER N
    REAL UTL,TA,SPACE,EMITP,OUTCON,EMITG,PTEMP,BETA,PI
    PI=3.141592654
C
C  FINDS THE TOP LOSS
C
    STEF=5.6697E-08
    F=(9/OUTCON-30/(OUTCON**2))*(TA/316.9)*(1+.091*N)
    D=EMITP+.0425*N*(1-EMITP)
    C=204.429*((COS(PI/2-COLALTR))**.252)/(SPACE**.24)
    UTL=(((N/(C/PTEMP*(((PTEMP-TA)/(N+F))**.252))+1/OUTCON))**(-1))
   *+(STEF*(PTEMP**2+TA**2)*(PTEMP+TA))/(1/D+(2*N+F-1)/EMITG-N)
    RETURN
    END
```

APPENDIX 6

Choice of Pentane Working Fluid

The choice of working fluid for the prime mover-collector combination was based on the desirability of certain physical and thermal characteristics. The desirable characteristics are:

- ready availability
- low cost
- boiling point just above atmospheric temperature
- non-flammable
- non toxic and non toxic derivatives
- high thermal conductivity
- low heat of vaporisation
- high vapour specific volume
- non corrosive
- non CFC

The need for a boiling point just above atmospheric temperature is so that condensation can be carried out at atmospheric temperature. It is also desirable to have the boiling point not too high above atmospheric temperature otherwise problems will be encountered with leakage of air into low pressure vapour areas ie. the condenser. The derivatives of the working fluid after contact with air and/or water should be non toxic. Given the current evidence about CFCs and their damage to the upper atmosphere, it would be unwise to use a CFC working fluid and ideally greenhouse gases should not be used either. A low heat of vaporisation and high specific gas volume will give a large volume change from the liquid to gaseous states with a small quantity of heat input.

Pentane fits all of the criteria except the non flammability, low cost and non toxicity. It rates moderately well in the low heat of vaporisation to high gas volume stakes, bettered mainly by CFCs. The price of pentane in New Zealand given in Chapter 3 was for the high purity Analar grade. Rao et al. (1976) reports that pentane is readily available and cheap in India. The general toxicity relates to that of petroleum: both have the same lower flash point, require similar cleaning and disposal after spillage and have the same handling precautions.

The majority of non CFC solar thermal water pumps use pentane (see Table 4) with no major problems having been reported.

The thermal and physical properties of n-pentane were taken from: Canjar et al. (1967), Gallant (1968), Jordan (1954), Lawrence et al. (1967), Osborn et al. (1974).